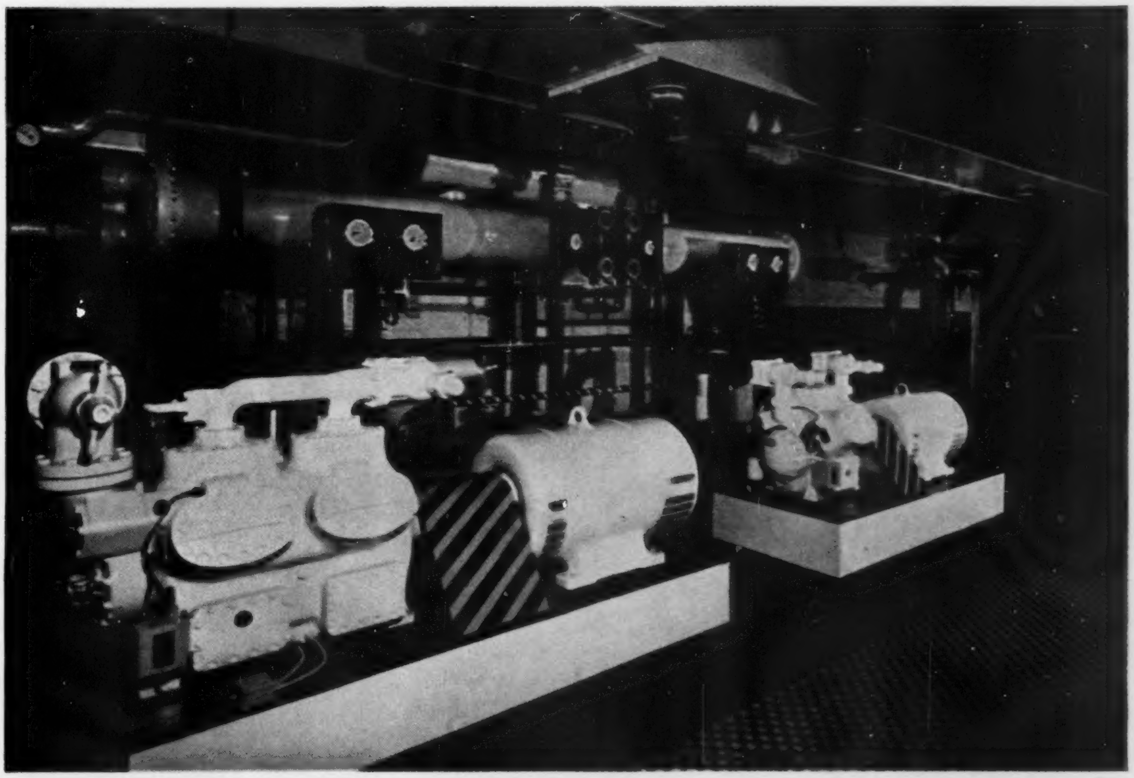
OFFICIAL PUBLICATION

ASHRAE JOURNAL

Heating ◆ Refrigerating ◆ Air Conditioning ◆ Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.



COLOR-CODING AND WELL-PLANNED EQUIPMENT LAYOUT FOR 165-TON AIR CONDITIONING INSTALLATION IN 6-STORY DEPARTMENT STORE HERE SERVES DUAL PURPOSE. SEE PAGE 78

NOVEMBER 1959

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VOL. 1 NO. 11

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Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

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Comment

BEING ALL THINGS TO ALL PEOPLE

There are those who lament that engineers are not joiners. It has been estimated that fewer than one half of the engineering graduates in this country belong to a society allied with their training and activities. There are protests, too, over the apparent indifference of the technical man who, having become a member of a professional society, fails to take advantage of opportunities to project himself into that organization's activities. We would that it were otherwise.

Yet, be it said, the character of professional engineering societies tends to a self-pigeonholing status which may be far narrower than the active interests of those who might participate to advantage within any one such organization. Accepting the fields of interest of ASHRAE to be primarily heating, refrigerating, air conditioning and ventilating, there are no less than seven organizations which the professional engineer might join. Should his interests be quite broad he must be torn between choices with resultant lack of concentration upon the possibilities of any one.

So how can a professional engineering society hope to serve the standing bareback rider who would guide three steeds at once? Quite a problem.

It is a tribute to the energy, intelligence and persistency of those members who direct professional engineering societies that the problem is solved so well, albeit in different ways at different times and for different groups.

Now firmly within a growth period, your merger-formed ASHRAE gives every promise of moving with notable success from a first brief interval of relative non-directionality into a firmly-paced program that will assure its leadership among its contemporaries.

Edward R Searles
Editor

Late news highlights

ASHRAE Committee for Exposition

Held under the auspices of ASHRAE in conjunction with its semiannual meeting in Dallas, February 1-4, the 2nd Southwest Heating and Air Conditioning Exposition will include exhibits from all parts of the U. S. President Arthur J. Hess is General Chairman of the Advisory Committee, which also includes the following members: C. Rollins Gardner, Chairman Arrangements Committee; F. H. Faust, member Board of Directors; Reg. F. Taylor, Past President; J. L. Collins, Pres. Shreveport Chapter; C. P. Houston, Pres. West Texas Chapter; H. D. McMillan, Pres. Houston Chapter; J. K. Mattox, Jr., Pres. Fort Worth Chapter; A. J. Mayers, Jr., Pres. Baton Rouge Chapter; J. J. Mays, Jr., Pres. Dallas Chapter; K. A. Monier, Pres. Alamo Chapter; T. A. Stokes, Pres. New Orleans Chapter; J. E. Tumulty, Pres. Northeastern Oklahoma Chapter; I. W. Wilke, Pres. Austin Chapter.

Plastics piping approved

Illinois has approved the use of plastics pipe and fittings for house or building service connections, for drainage and vent piping within the building and in cold water piping systems outside the building walls or foundations, as set forth in a new state plumbing code released in October.

Cryogenics series

In order to keep abreast of the increasing number of articles relating to new developments in various branches of cryogenics appearing in numerous journals and to make workers in one field of low temperature technology familiar with techniques advanced in others, a series of volumes has been planned to consist of from eight to ten critical reviews of selected topics in each volume by authors from any country in which significant work is being carried out. Production, maintenance and measurement of low temperatures and their practical application and techniques used in basic research will be covered. Now available is *Progress in Cryogenics*, Volume I, from the Academic Press Inc., 111 Fifth Avenue, New York, N. Y., price is $11.

Indiana scholarship

A scholarship fund at Purdue University has been established by the Central Indiana Chapter of ASHRAE to award scholarships to students enrolled in mechanical engineering or engineering sciences allied with the arts of heating, ventilating, air conditioning and refrigerating at the end of their second or third year in the University, based upon scholarship, character and financial need. Set up with the Chapter's surplus funds for last year, the scholarship can be perpetuated by donations from anyone.

Research conference

How to develop better means of communication of ideas in the research fields in order to stimulate young scientists and engineers to undertake projects of potential significance will be discussed at the conference on Research Goals to be held at Worcester Polytechnic Institute, Worcester, Mass., December 3 and 4, under a grant from the National Science Foundation. Fifteen scientific and engineering societies, together with college deans and presidents and directors of research in industry and government, will participate in the conference.

First in Britain

First Industrial and Commercial Refrigeration Exhibition to be held in Great Britain, sponsored by World Refrigeration, will take place in London, February 15-17, 1960.

Comprehensive Directory

Building Science Directory, published by the Building Research Institute, National Academy of Sciences, National Research Council, contains alphabetical reference lists of nearly 500 associations and societies of the building industry in the U.S.; more than 200 private research and testing facilities for the building industry, including fields of specialization for each organization; nearly 100 colleges and universities doing research on building in the U.S., listing the types of projects. Additionally, there are detailed data pages on 200 selected associations, plus an annual index by name and subject matter. Supplementary data pages are issued quarterly. The Directory may be obtained from BRI, 2101 Constitution Avenue, Washington 25, D. C. Cost is $14 for initial volume and $5 for an annual subscription to begin in 1960.

Third edition Extensive revisions and additions, covering advances in engineering and new devices for industrial environment control, bring up-to-date the third edition of *Design of Industrial Exhaust Systems* by John L. Alden. Taking into account the increasing appreciation of sound engineering principles relating to control and cleaning of industrial air, this edition describes such developments as fog filters, reverse flow filters, venturi scrubbers and multiple miniature cyclones. Subjects covered include: fluid flow, hoods of various types with their characteristics and specifications for different operations, duct system design, dust separators with features of numerous types, pneumatic conveyors, fans, system detail and planning, field measurements and their interpretation. Industrial Press, 93 Worth Street, New York, N. Y., $6.

NSPE award An Industrial Professional Development Award will be presented annually by the National Society of Professional Engineers, beginning with its annual meeting in June 1960, to the industrial employer of engineering personnel which has made an outstanding contribution to the advancement and improvement of the engineering profession through its employment practices. According to Harold A. Mosher, NSPE president, the award has a two-fold purpose – "to give adequate recognition to the firm which has demonstrated the application of forward-looking engineering employment practices and to encourage industrial employers of engineering personnel to adopt progressive engineering employment practices in accord with established professional standards."

Ice sales National Ice Association reports in its annual survey that ice sales for 1958 exceeded 22 million tons, representing slightly over $207 million. These figures were compiled only from reporting companies; more complete data will probably raise the total.

AAAS reports Activities of engineering interest of the American Association for the Advancement of Science are available in a report which covers the latter part of 1958 through the fall of 1959. During the December 1958 meeting a four-session conference on National and International Aspects of Systems of Units was held, with 28 leaders of industry, education and government from six countries participating; proceedings will be published as a Symposium volume.

Reprints Available in pocket sizes are reprint editions of Refrigeration Manuals, Volume I, "Refrigeration Educational Course" and Volume II, "Common-Sense Talks on Commercial Refrigeration." These books, which consist of material published originally as a series of articles in Modern Refrigeration, may be obtained from Refrigeration Press Ltd., 131 Great Suffolk Street, London S.E. 1, England. Price of each volume is 15 shillings net.

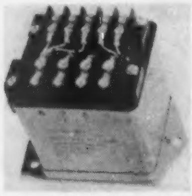
Porosity of plated coatings Since the protective value of a plated coating on metal is dependent to a large extent on the exclusion of pores and other flaws in the coating, the National Bureau of Standards, Washington 25, D. C., undertook an investigation of the causes, nature and effect of porosity in plated coatings. Several aspects of porosity were studied, including physical and chemical methods of detecting pores and the influence of gas on pore formation. Results of the study, which have been published in a summary technical report, indicate that flawless coatings are especially important in high temperature applications, such as jet engines, because metals must resist sharp changes in environment.

Accomplishments to be reviewed Speakers and subjects at the 1959 convention of the National Warm Air Heating and Air Conditioning Association, to be held in St. Louis, Mo., on December 2-4, will focus attention on "a year of extraordinary accomplishment," according to Tom Byrd, president of the Association. Electric heating will be discussed by John Norris, president of Lennox Industries and Edwin O. George, vice president of the Detroit Edison Company. Research activities and findings during the past year will be summarized by Keith T. Davis, chairman of the Research Advisory Council of the Association and Edward J. Brown, head of the Association's Research Staff at the University of Illinois. Educational activities and accomplishments of the Association during 1959, including the impact of new instruction materials, will be reported upon by Professor Lorin G. Miller, chairman of the Application Engineering Council.

PARTS AND PRODUCTS

RELAY DEVICE

Consisting of a magnetic relay, two transistor amplifiers, a diode network for signal separation and a voltage regulated power supply, the direct monitoring over-temperature relay device shown may be used in a wide range of ac applications. Although specifically designed to protect 3-phase ac motors, this relay is suitable for use wherever over-temperature is a problem. Power supply ratings are 110, 208/220, 440 or 550 volt, 60 cycle. Contacts are the pilot duty type, 600 volt ac maximum.



Cutler-Hammer, Inc., 501 N. 12th St., Milwaukee, Wis.

GAS-FIRED CONDITIONER

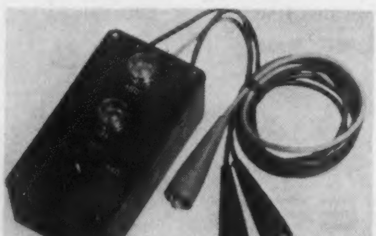
As a remote-type unit, the Oasis gas-fired air conditioner offers complete installation flexibility. Installed wherever convenient in an outside location, the cooling unit supplies chilled water to the cooling coil which is installed inside on the furnace for a year round system or to the fan-coil blower unit where the system is for cooling alone.
Day and Night Manufacturing Company, P. O. Box 2222, La Puente, Calif.

FURNACE FILTER GAUGE

Cited as substantially increasing heating efficiency for commercial and home users, this gauge is designed for use with Pollenex furnace filters to provide a visual check on the filter's efficiency and signal when cleaning is required.
Associated Mills, Inc., 303 W. Monroe St., Chicago 6, Ill.

PHASE SEQUENCE INDICATOR

Model 40, a portable instrument for indicating phase sequence in three-



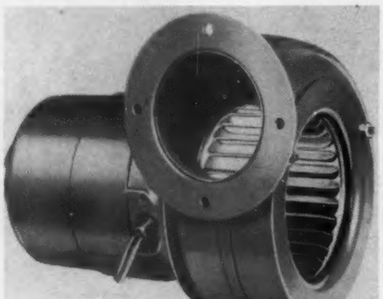
phase power systems, is for 60 cycle power and has a switch for adjustment to 120, 240 and 480 volt. Three insulated test leads are connected at random to the three-phase conductors. When the lamp marked "ABC" glows, the sequence is as designated by the test leads. If the lamp marked "CBA" glows, the phase sequence is the reverse of the lead designation. For use with 400 cycle power systems, Model 44 is available.
Associated Research, Inc., 3777 W. Belmont Ave., Chicago 18, Ill.

ICE MACHINE CLEANER

Packaged in plastics envelopes containing enough for a single treatment, a crystalline soluble chemical dissolves mineral deposits and corrosion that form in ice cubers and similar equipment. Contents of the package are poured into the make-up water sump and recirculated until the system is thoroughly cleaned of all deposits.
Stiles-Karlsonite Corporation, 1550 Grand Ave., Waukegan, Ill.

CENTRIFUGAL BLOWER

Recommended for applications where nominal ambient temperature conditions can be anticipated, Air Unit No. 89C21, here shown, is designed for commercial applications, being especially well-suited for cooling electronic tubes, ventilating small elec-



tronic equipment cabinets and cooling hot spots. Having a 1/40-hp shaded-pole electric motor of the unit bearing type not requiring oiling, the blower operates at 3000 rpm using 115 or 230 volt, single-phase, 60/50 cycle current. Operating at rated speed, it delivers 106 cfm of air at zero static pressure and 20 cfm of air at one in. w.g. static pressure.
American Radiator and Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

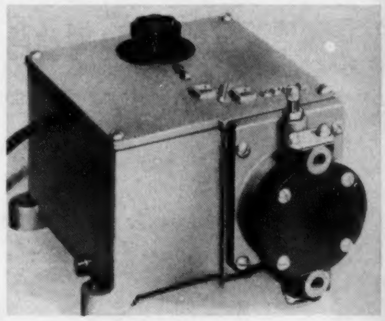
FOUR HEATING UNITS

Two oil-fired and two gas-fired furnaces, specifically designed for use in lower heat-loss homes or in multiple-unit, zone heating systems, have been added to this line of residential heating units.

All four models are of a compact design for closet installation in basementless homes with cement slab or crawl space construction. Models OBC 70-HR and GBC 80-HR (oil- and gas-fired, respectively) are counterflow designs for perimeter heat distribution systems, while Models OBC 70-H and GBC 80-H are designed for overhead duct systems.
General Motors Corporation, Delco Appliance Div, Rochester 1, N. Y.

DIAPHRAGM PUMP

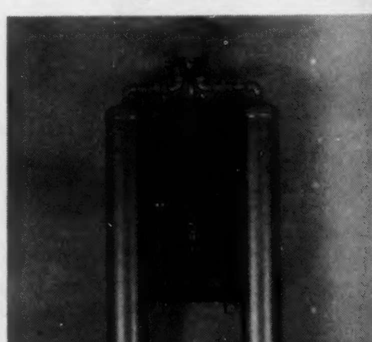
Added to this line of controlled capacity Chemical Metering Pumps is a diaphragm pump designed for positive displacement metering of a wide range of corrosive and non-corrosive



liquids. Operating against back pressures as high as 500 psi, the pump has four flow ranges available up to the maximum of 2 5 gph. Construction includes built-in overpressure relief valve, with a hydraulically-balanced Teflon diaphragm cited as eliminating flexing fatigue and distortion of flow rate. The pump is available in multihead construction and for automatic proportioning from pneumatic or electric impulse signal.
Clark-Cooper Company, 315 Market St., Palmyra, N. J.

COMPRESSED AIR DRYERS

Two sizes of dryers, for use in connection with pneumatically operated

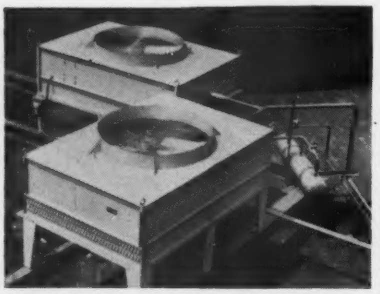


control systems and any other applications where a small quantity of air, from one to 25 scfm, at inlet pressures up to 125 psig and 90 to 100 F inlet

temperature, is used, have been introduced by this manufacturer. Fully automatic, the dryer is simply constructed, having only two moving parts, a timer and solenoid type 4-way valve. Desiccant towers are of the throw-away type and easily replaced. Built-in dust filters are cited as assuring clean, dry air at dew points down to −60 F and below.
Desomatic Products, Inc., 1109 W. Broad St., Falls Church, Va.

VERTICAL FLOW AIR-COOLED CONDENSERS

Ranging in size from three to 100 ton, these units are cited as offering many advantages, such as horizontal mounting of fan and coils for a lower silhouette (see cut), upward draft re-

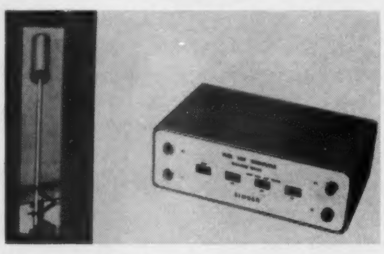


sulting in noise being directed away from nearby structures, embossed fins creating better air wash and lowering air film resistance and mounting on legs to reduce hot roof air which might be drawn in were physical clearance not provided.
Halstead and Mitchell, Zelienople, Pa.

ANALOG COMPUTER

Automatically indicating average fuel oil consumption continuously, under all weather conditions, and relaying the information to a control panel, the Fuel Use Computer was developed for fuel oil dealers to provide a means of determining the amount of fuel oil their customers are consuming.

Key components of the computer are a cylinder-shaped analog roof-top unit and a recording console the size



of a portable typewriter. Instruments inside the roof unit create an electrical equivalent, or analog condition, of heated buildings exposed to weather. As weather changes, the analog unit adjusts its energy output, this change being relayed to the recording console where it is measured in digital readings. Three digital readings on the console indicate the amount: the first is for installations with low hot water requirements, the second for average requirements, the third for high. A fourth register indicates fuel use for heating only.
HRB-Singer, Inc., 90 Schureman St., New Brunswick, N. J.

STRAINER-FILTER-DRIER

Protection of refrigeration systems against acids, moisture and dirt is cited for A-P Model 410 Trap-Dri, a combined strainer, filter and drier featuring a blend of silica gel and molecular sieve desiccants that absorbs water and acid without release of substance to the refrigerant circuit.

A honeycomb filter tube constructed by diagonally winding processed coarse cotton yarn of soft texture around an open metal supporting core to form diamond-shaped filtering tunnels, the unit is offered with solder or flare type connections, 1/3 to 7½-ton capacities.
Controls Company of America, Heating and Air Conditioning Controls Div, Milwaukee, Wisc.

COMPACT COOLING UNIT

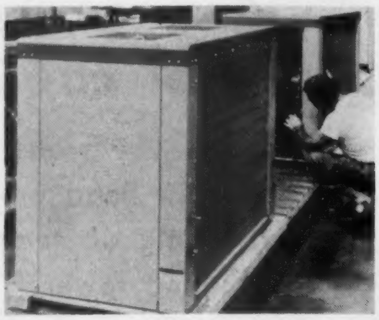
Designed with compactness in mind, the Top-Aire, shown below, is flat and thin, leaving extra shelf space avail-

able in reach-ins, back bars and undercounter cabinets. Other features are drain fitting angled so it can be run through back or bottom of cabinet, space for expansion valve inside unit and knockouts on top plus openings in rear for bringing out connections. Capacities, based on entering air temperatures, are 1000, 1300 and 1700 Btu/hr at 10 deg T.D.
Bohn Aluminum and Brass Corporation, Betz Div, Danville, Ill.

5-TON HEAT PUMP

Added to a line consisting of 3- and 7½-ton units is an air-to-air 5-ton heat pump, Model RHP-53, equipped with a control system that senses the outdoor air temperature and regulates indoor temperature accordingly. A split-system heat pump, the unit has separate indoor and outdoor sections, and is adaptable to both residential and commercial applications. Shown

are the first outdoor sections in the final stages of manufacture. Designed for flexibility of installation, the indoor section can be assembled for vertical, counterflow or horizontal flow of air and can be installed in a

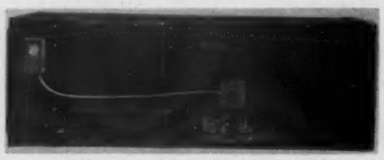


small closet, attic, crawl space or garage, in addition to standard locations such as basements and utility rooms.

Supplementary duct heaters are available as optional accessories in steps of 6 kw up to 24 kw total heat. A minimum of 6 kw is required for defrost. Thermal cutout switches in these heaters prevent overloading.
Westinghouse Electric Corporation, Air Conditioning Div, P. O. Box 510, Staunton, Va.

FOUR GAS-FIRED FURNACES

Available in four sizes, 80,000, 100,000, 120,000 and 140,000 Btu input, the new line of Series YY Luxaire Gas Horizontal Furnaces contains models more compact than those they replace.



Units feature low height for installations in low attic or crawl spaces.

For cooling, standard-equipment blowers which provide additional air deliveries compatible with the sizes of cooling coils with which these furnaces are usually installed are optionally available.
C. A. Olsen Manufacturing Company, Elyria, Ohio.

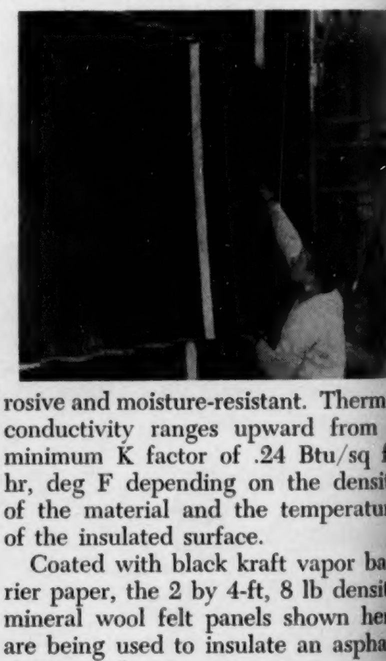
CONTROLLER-POSITIONER

Combining in one valve-mounted unit the functions of an indicating pneumatic controller for temperature or pressure and a valve positioner which amplifies air power to provide accurate and rapid positioning of a pneumatically-operated control valve, is the "Pilot-Positioner." It can be mounted by the valve manufacturer on most makes of diaphragm motor valves or pneumatic operators, and requires only one air supply.

Pressure elements cover the range of 30 in. hg vacuum to 10,000 psi (also suitable for liquid level control) and filled-system thermometer elements cover the range of –350 to 1000 F. For use with pneumatic transmitters, the unit can be furnished as a pneumatic receiver, indicating the variable at the valve and permitting additional indication or recording at a remote panel board.
United States Gauge Div of American Machine and Metals, Inc., Sellersville, Pennsylvania.

FELT INSULATION

Available in densities from 3 to 8 lb/cu ft, Spun Felts are designed for use in high temperature industrial equipment, heated appliances and refrigeration systems. Made from spun mineral fibers with a phenolic resin binder, the felts can be used to insulate surfaces having temperatures from below zero to 450 F. If mechanically supported and enclosed they may be used satisfactorily up to 600 F. Lightweight and resilient, the insulation is incombustible, non-cor-



rosive and moisture-resistant. Thermal conductivity ranges upward from a minimum K factor of .24 Btu/sq ft, hr, deg F depending on the density of the material and the temperature of the insulated surface.

Coated with black kraft vapor barrier paper, the 2 by 4-ft, 8 lb density mineral wool felt panels shown here are being used to insulate an asphalt storage tank.

Felt sizes are 24, 30, 36 or 48 in. long by 12 or 24 in. wide, and come in thicknesses of from one to four in. in ½-in. increments.
Baldwin-Ehret-Hill, Inc., 500 Breunig Ave., Trenton 2, N. J.

WATER TREATMENT

Complete in one package, EZ Treat is a formulation to prevent scale, corrosion and slime formation in cooling towers and evaporative condensers of 5 to 50-ton capacity. Supplied in specially designed feeder-cans which allow a gradual, automatic dis-


PEOPLE PUT THE PLUS IN VIKING TUBING
Viking's position as one of the nation's leading suppliers of thin-wall copper tubing has been achieved through the skill and dependability of the *people* at Viking.
Since Viking specializes in the manufacture of copper tubing for the refrigeration and air conditioning industry, Viking people are experts in the field. They know the importance of rigid adherence to the highest standards of temper, tolerance and uniformity to meet the customer's most exacting needs and specifications. They gear themselves to the needs of customers . . . adjusting and revising production and delivery schedules to conform to standard or emergency demands.
As a result, Viking quality and Viking service add up to a valuable *plus* for manufacturers of air conditioning and refrigeration units and coils.
The *people* at Viking will gladly discuss your copper tube requirements with you . . . you'll find the *plus* pays dividends.
VIKING
COPPER TUBE CO
CLEVELAND 10, OHIO
PRECISION DRAWN SEAMLESS COPPER TUBE

pensing of the chemical treatments, the product requires no special feeding apparatus or other devices.

Chemicals in the form of tablets attached to the cover of each feeder-can control the growth of slime and other organic elements in air conditioning equipment when removed and thrown into the water. Feeder-cans are located in the pans of the equipment so that, through the natural circulation of the water, the chemicals for treating scale and corrosion are gradually and continually dispensed into the system.

Metropolitan Refining Company, Inc., 50-23 23rd St., Long Island City 1, New York.

PRESSURE CONTROL VALVE

Designed to hold pressures in tanks within 0.1 psi despite varying inlet pressures, this regulator valve may be adapted to a variety of sizes and pressure settings. It operates from zero to 300 psig, maintaining full flow within 0.2 psi above cracking pressure, and will function efficiently in a temperature range from −65 to 480 F.

Besler Corporation, 4053 Harlan St., Emeryville, Oakland 8, Calif.

WASHABLE URETHANE FILTER

Offering several advantages from an engineering standpoint, this filter's most important feature for the owner is its easy maintenance. When it becomes dirty, the filter may be washed, wrung dry and re-installed in the unit. No replacement is necessary, as in the case of a fiber-glass filter, nor is spraying with oil required, as with aluminum mesh filters.

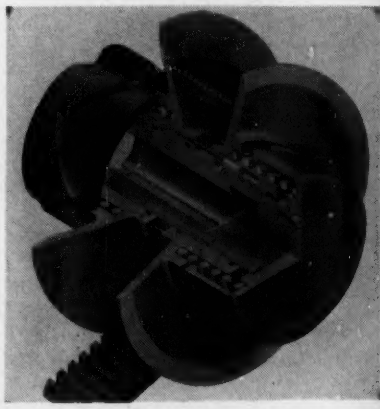
For filtering purposes, a special kind of urethane foam, ScottFoam, is used, differing from conventional urethane foam in that it does not contain the thin membranes which usually enclose the gas bubbles in the material. Without these membranes, the foam is a fully skeletal structure, offering little resistance to airflow, a characteristic important to the engineer concerned with pressure drops across the filtering medium.

Allied Chemical Corporation, 40 Rector St., New York 6, N. Y.

MOTION-CONTROL SHEAVE

Designed to hold a constant driven speed during changing torque loads, the MCS variable-speed sheave incorporates a lubrication system cited as eliminating fretting corrosion, freezing and sticking. Cam pressure is exerted only when the load requires it, and power is transmitted equally to both movable flanges through a series of torsionally resilient rubber cam followers.

Two sizes are offered: MCS-10-W, for motors of 7½- to 15-hp, has a



minimum pitch diam of 5.0 in. and a maximum pitch diam of 10 in.; MCS-12-W, for motors of 15- to 20-hp, has a minimum pitch diam of 6.0 in. and a maximum pitch diam of 12 in.

T. B. Wood's Sons Company, Chambersburg, Pa.

AIR CONDITIONER LINE USES GERMICIDAL FILTERS

Fan-coil room air conditioners in this manufacturer's line are now available with the germicidal, replacement-type air filter manufactured by Fram Corporation. Also available with these filters are the wall-mounted line of induction circulators and flexular series induction circulators.

Worthington Corporation, Harrison, New Jersey.

THREAD TAPE

Made of chemically inert Teflon, TTP Thread Tape provides leakproof threaded pipe joints for unplasticized polyvinyl chloride, stainless steel, aluminum and other corrosion-resistant piping. Unaffected by corrosive materials, the tape is self-lubricating and prevents joints freezing.

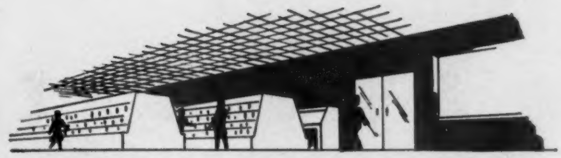
Tube Turns Plastics, Inc., Louisville 11, Ky.

POLYETHYLENE TUBING

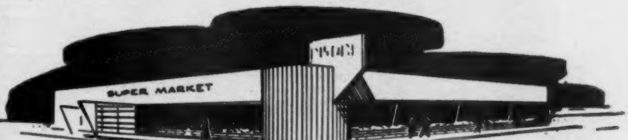
Specially designed for use in ice rinks featuring machine-made ice, this plastics tubing is a high-density, high-strength extruded product with a wall thickness only half that of other ice rink tubing, and is cited as allowing faster cooling and eliminating need for extremely low brine temperatures.

A major problem in plastics ice rink tubing has been contraction in length as it cools from installation to operating temperature. This polyethylene tubing contracts only seven in. over 185-ft length as the refrigerant it carries drops to 20 F.

The McQuay "AL" AIRCON air cooled condenser provides the ultimate in flexibility by combining refrigerant condensing with heating and ventilating. You can utilize condenser heat for heating, and this AIRCON may also be used for exhaust ventilation, saving both time and money in installation, maintenance and operating costs.



INSTALL IT INDOORS



INSTALL IT OUTDOORS

"AL" AIRCON is compactly designed, and because of a very low silhouette, will not detract from the appearance of a building when roof mounted . . . the largest model is less than 4½ feet high, unnoticeable from the street.

"AL" AIRCON condensers are available in 8 different fan discharge arrangements and in 16 different unit arrangements . . . all standard. Each condenser coil is factory circuited in single or multiple sections to meet specific requirements of either air conditioning or refrigeration applications. Contact your nearest McQuay representative, or write McQuay, Inc., 1606 Broadway Street N. E., Minneapolis 13, Minnesota.

utilize condenser heat with this low silhouette blower type AIR COOLED AIRCON

McQuay Means Quality

McQuay INC.

AIR CONDITIONING • HEATING • REFRIGERATION

MAKE THE SEASONS COME TO YOU

Produced in 185-ft coils in a nominal 1-in. size with an outside diam of 1.15 in. and an inside diam of 1.05 in., this tubing eliminates the need for joints, because of its long length. Minimum burst pressure is 370 psi at 20 F, 245 psi at 75 F and 190 psi at 100 F. Maximum working pressure for an anticipated service life of 50 yr is 135 psi at 20 F, 60 psi at 75 F and 30 psi at 100 F.
United States Steel Corporation, National Tube Div, 525 William Penn Pl., Pittsburgh 30, Pa.

ONE-PIECE AIR CHAMBER

Intended to remove hammer and chatter from water systems is a one-piece prefabricated air chamber that can be installed directly into each supply line in the system by making only one soldered joint. Leakproof, Tap/Trap is available in 6- and 12-in. lengths.
Wolverine Tube, Div of Calumet and Hecla Inc., 17200 Southfield Rd., Allen Park, Mich.

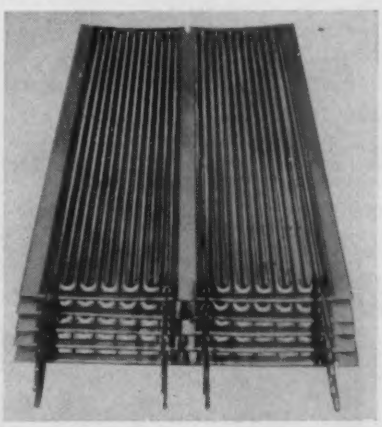
7½-IN. HIGH BASEBOARD

Featuring a low overall height of 7½-in. and a depth of only 2⅝-in., Model F hydronic baseboard is designed to meet the modern trends in home construction. An important innovation is box-type aluminum fins cited as being sturdy enough to support the weight of a man. The box design of the fins increases heat transfer surface and directs the draw of air by providing individual flues through-out the length of the element.

Another series, Model RL, has all of the design advantages of the F series, but its greater capacity requires an over-all height of 9⅝ in.
Fedders Corporation, 58-01 Grand Ave., Maspeth, N. Y.

REFRIGERATION PLATE

Designed to become an integral part of the inner tank in a bulk milk cooler, the Platecoil unit consists of



two pieces of metal, one flat and the other embossed, seam-welded together. Of 14 gage stainless steel, the flat piece forms the tank bottom; the other plate is of 20 gage stainless steel and embossed in a serpentine pattern, with the embossed area on the outside of the inner tank. The raised part of the embossed plate forms channels through which the refrigerant passes, with no additional tubing required.
Tranter Manufacturing, Inc., 735 E. Hazel St., Lansing 9, Mich.

NONCORROSIVE NOZZLES

Hollow cone spray nozzles, injection molded of Tenite Butyrate, provide noncorrosive spray heads for air

washers, cooling towers, spray ponds and evaporative condensers. Spraying begins at ½ psi because of the design and the low inherent friction of the material. Being nonporous, this material resists build-up of lime and other water mineral deposits.

Seven capacities (0.8 to 7 gpm) and three F.P.T. sizes (⅜, ½ and ¾ in.) are available.
Austin Manufacturing Corporation, 305 Perry Brooks Bldg., Austin 1, Tex.

48-FRAME MOTOR

Length of the 48-frame fractional hp motor, designed primarily for the heating and air conditioning industries, is up to ½ in. shorter than the previous design, to match recent trends towards greater compactness. The new designs are for shaded-pole motors up to ¼ hp and permanent-split capacitor motors up to 1/3 hp.

Motors are available either with line leads or a terminal board, equipped with spade connectors, mounted over the vent holes in the frame. Units are designed for versatility of mounting: rigid or resilient, band, end-mount with single- or double-extended studs or halo mount.
Westinghouse Electric Corporation, P. O. Box 2099, Pittsburgh 30, Pa.

PIPE COVERING

Widely used in slab and block form for a variety of low temperature insulating requirements in the residential and commercial building fields, this expanded polystyrene is offered as pipe covering for a wide range of design conditions. Characteristics cited for the material are low K factor, odorless, non-toxic, moisture-resistant, compression strength of 16 to 20 psi, condensation and drip prevention, and stopping of heat gain.

Three standard sizes in both regu-

lar or self-extinguishing types are available: 1 in. for temperatures above freezing, 1½ in. for temperatures of 32 F to zero and 2 in. for temperatures of zero to –30 F. Extra thicknesses are available for more severe conditions.
United Cork Companies, Kearny, N. J.

GAUGE MOUNT

Lengthening gauge life by damping mechanical vibration and gauge pointer jiggle, this mount is designed for bottom-inlet gauges weighing up to two lb and can be used for any gas or fluid that does not attack carbon steel. It is especially applicable for use on reciprocating compressors and piping systems in which the gas or fluid flow has pulsating characteristics.

A rubber damper isolates the pressure gauge from the compressor or piping system, avoiding a rigid connection between the two and enabling the gauge to stand motionless. Gas pressure is transmitted to the gauge through a small-bore, stainless steel, spiral-wound tube so that only a smooth, even pressure reaches it.
Vilter Manufacturing Company, Milwaukee 7, Wisc.

POTTING COMPOUND

Permitting visual and instrument checking of individual parts within a potted assembly, Dielectric Gel, a transparent silicone potting material, cures in place to form a resilient, protective mass that retains its dielectric properties and moisture resistance over a temperature range of below –60 to 200 C. In addition to filling and potting applications, this material holds promise as a damping material and can be used also as an impregnant for capacitors, magnetic amplifiers and similar devices.
Dow Corning Corporation, Midland, Mich.

1960 CONDITIONER LINE

Comprised of 38 models, this manufacturer's 1960 line of window air conditioners includes three one-hp Minute Mount models, one with a capacity of 7000 Btu for installation in casement windows and two, of 7000 and 8000 Btu capacities, for installation in standard double-hung type windows.

Highest capacity unit in the line is

the 22C-3 with 19,000 Btu. This air conditioner is a five-row evaporator and five-row condenser type with a twin cylinder compressor and a three-speed fan motor.
Fedders Corporation, 58-01 Grand Ave., Maspeth 78, N. Y.

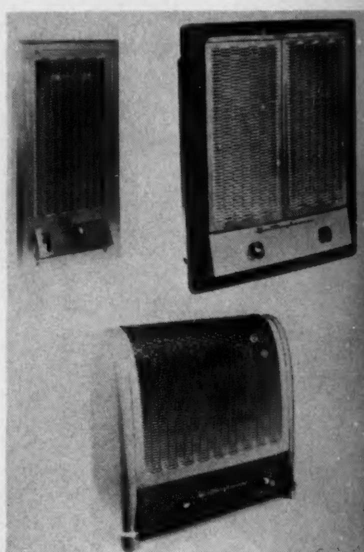
ELECTRIC HEATERS

Three completely redesigned models of residential electric heating equipment have been introduced, consisting of wall, floor and bathroom models. Each of these units incorporates a new sloped control panel that is easier to read and operate than that of previous models.

Wall models are available in seven sizes from 1000 to 4000 watt operating at 240 volt. A bellow-actuated thermostat maintains desired temperature within two deg, and is adjustable from 55 to 85 F.

Two sizes, 1250 or 1500 watt at 120 or 24 volt, of bathroom models are available, with or without adjustable heat control.

Available in 2000-, 3000- or 4000-watt sizes, floor models are suited for use in hard-to-heat areas such as recreation rooms or work areas. They can easily be moved from one room to



another and require no installation other than plugging into a 240-volt outlet. Adjustable temperature control is built-in.

Shown in the cut are the bathroom and wall models at upper left and right, respectively, and floor model at the bottom.
Westinghouse Electric Corporation, P. O. Box 868, Pittsburgh 30, Pa.

ALTITUDE CHAMBER

First in a line of environmental chambers that will test products and materials while undergoing radiation exposure, this three cu ft test chamber

Designed to give you *Double Protection!*

WAGNER TYPE DP MOTORS

PROTECTED against corrosion...

PROTECTED against splashing liquids...

Wagner Type DP Motors provide *double protection* that means longer life—more versatility of application. Rugged cast iron frames and endplates are highly resistant to corrosion. Dripproof enclosures are so well designed that these motors can handle many applications that formerly required splashproof motors. These motors pack ample power into little space, are light in weight and are easy to maintain.

SLEEVE BEARING MODELS AVAILABLE. The entire line of ratings is available with ball bearing construction, or with steel-backed, babbitt-lined sleeve bearings of high load carrying capacity that provide quieter operation.

Let a Wagner Sales Engineer show you how these motors can be applied to your needs. Call the nearest branch office or write for Wagner Bulletin MU-223.

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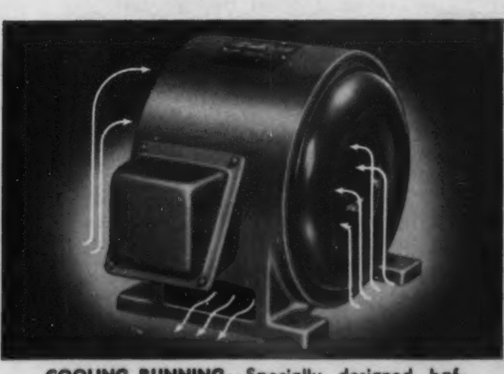
DOUBLY PROTECTED—Air intakes and outlets are positioned to provide complete protection against dripping or splashing liquids. Rugged cast iron frames protect against rough handling and corrosion.



COOLING RUNNING—Specially designed baffles, which protect the stator windings, direct a cooling stream of air through the motor to effectively cool the motor—add to motor life.

CAN BE RE-LUBRICATED—Original factory lubrication will last for years in normal service—but grease plugs are provided to permit re-lubrication that adds years to motor life under severe conditions.

has an adjustable temperature range from –80 to 180 F. Altitude can be controlled from sea level to 100,000 ft, accomplished in 15 min time. A steam vapor generator provides up to 100% relative humidity in the chamber.

Radiation is accomplished by a 1-in. diam non-metallic tube penetrating

through the chamber and extending 6 in. on each side. A capsule of radioactive material is inserted in the tube for the test, but the chamber can be used with or without the capsule in the tube.

Cincinnati Sub Zero Products, 3932 Reading Rd., Cincinnati 29, Ohio.

RECONDITIONING COMPOUNDS

Two compounds for reconditioning heating systems are offered, one removing accumulated boiler rust and sludge, the other sealing leaks without shutdowns.

Heat-Pep Cleaner removes rust and sludge in old boilers and loosens core sand, oil and dirt common to new ones, simultaneously coating boiler walls to protect them from further accumulations of impurities.

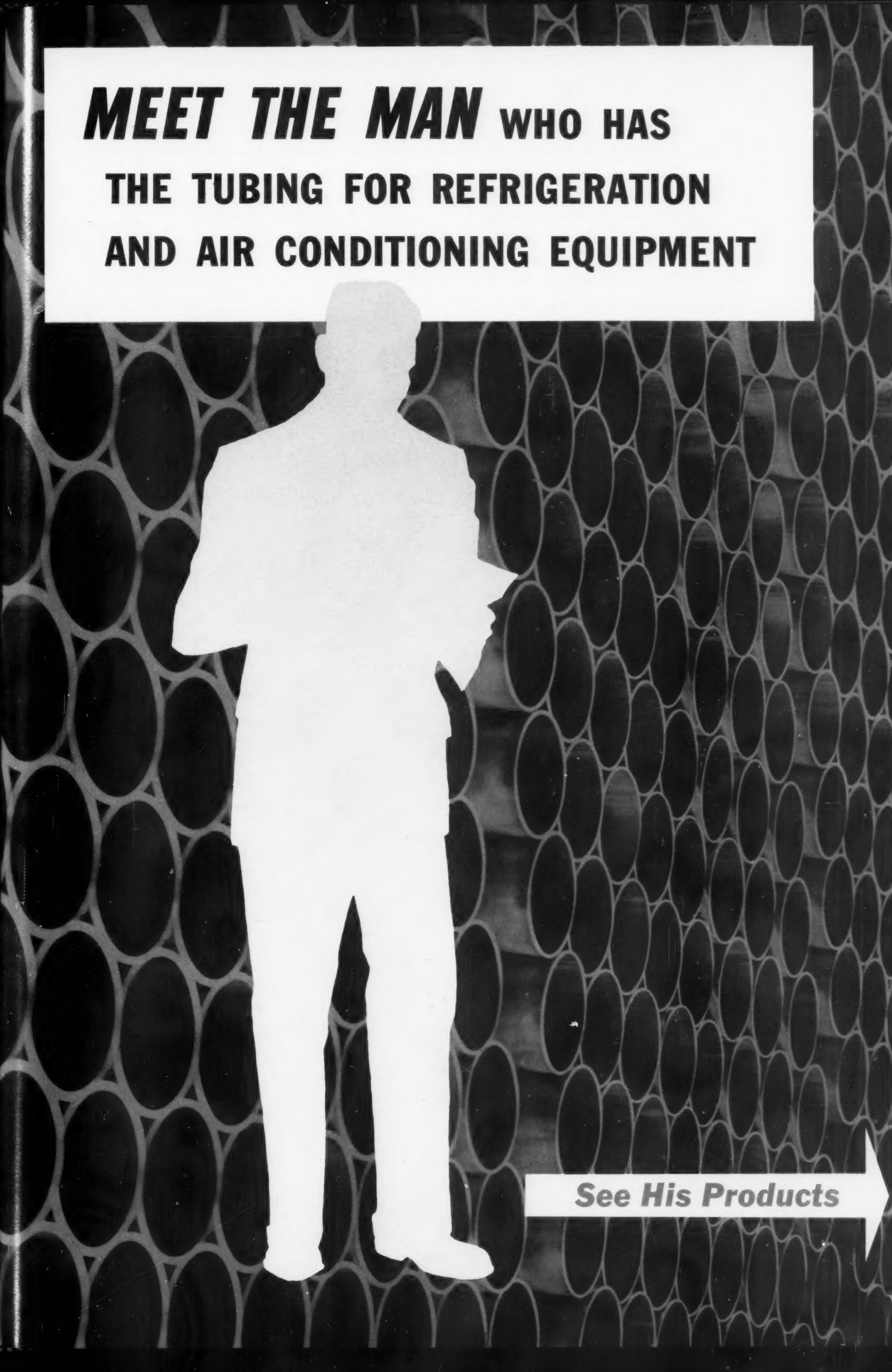
Heat-Pep Sealing Compound mixes with system water, and is forced by boiler pressure to leaking parts, forming a permanent bond that becomes an actual part of the boiler. This seal is not affected by expansion or contraction.

J. A. Sexauer Manufacturing Company, 2303-05 Third Ave., New York 51, N. Y.

PORTABLE EVACUATING AND CHARGING STATION

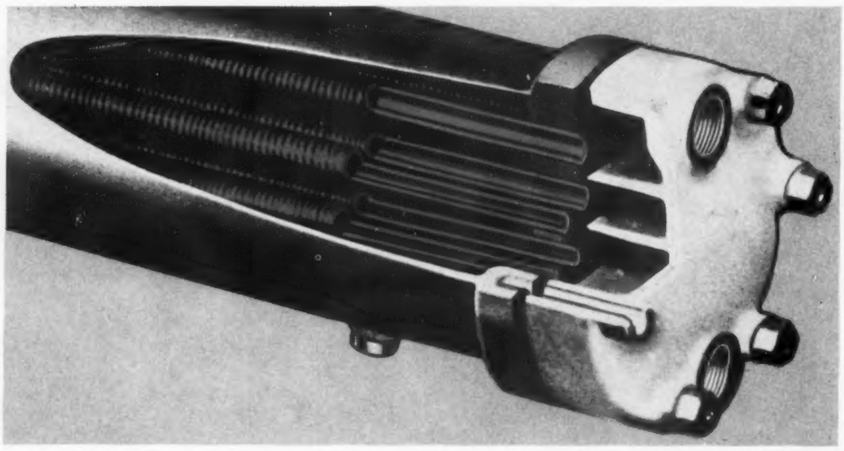
Universal for all automotive air conditioning systems, the station features a cylinder that measures the correct amount of Refrigerant-12 to be used for each charging service operation, and charges with accuracy anywhere between 60 and 110 F. Components

(*Continued on page 44*)


MEET THE MAN WHO HAS
THE TUBING FOR REFRIGERATION
AND AIR CONDITIONING EQUIPMENT
See His Products

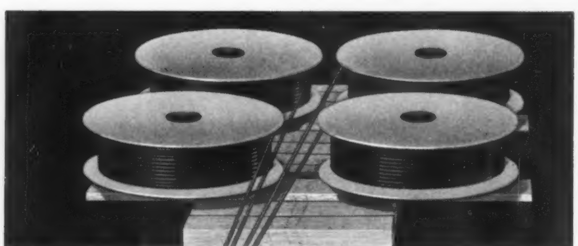
THESE WOLVERINE PRODUCTS CAN HELP YOUR COMPANY DO A BETTER JOB

Wolverine Trufin is also available with high fins—can be readily fabricated to meet your needs.

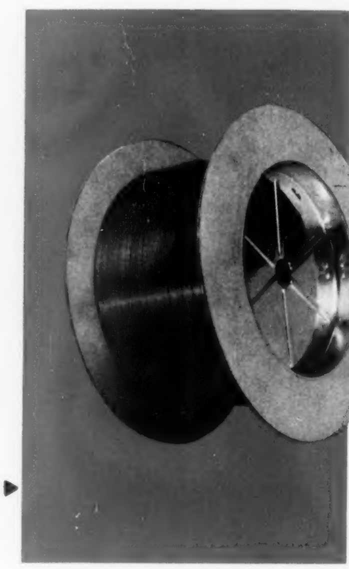


Wolverine Trufin® Type S/T—the integrally finned condenser tube for shell and tube condensers.

Wolverine Capilator® for precision metering control.



Wolverine level-wound coils are ideal for feeding automatic production equipment.



Wolverine control tube is also available in handy level-wound coils.

Wolverine's aluminum refrigeration assembly has aluminum to copper connections.

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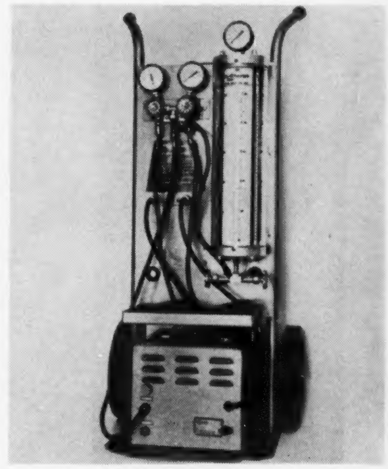
(*Continued from page* 40)

of the unit shown include all necessary gauges, fittings, high-pressure hoses, "Dial-a-Charge" cylinder, vacuum pump and welded brackets to



accommodate a 25 lb refrigerant drum, all mounted on a portable cart with roller bearing wheels.
Vehicle Products Div, Robinair Manufacturing Corporation, Montpelier, Ohio.

STEAM TRAPS

A combination inverted bucket and thermostatic steam trap, Heat-Kwik traps drain condensed moisture and vent air faster than "blast" traps. When steam is first turned on the bellows is wide open and the inverted bucket is down with its valve open, permitting condensation and air to discharge. Upon steam reaching the trap, the bellows expands and closes its valve and the bucket valve is closed. When moderate amounts of condensation or air reach the trap again, the bucket drops, opening the valve so that they can be discharged.
V. D. Anderson Company, 1935 W. 96th St., Cleveland 2, Ohio.

VENT EXHAUSTER

Unit heaters are cited as being made suitable for heating of multiple-story buildings with the development of a motorized vent exhauster. Requiring only a short length of standard 4-in. pipe, the device vents exhaust gases from unit heaters directly to the outdoors through the side wall of the building. Powered by a permanently lubricated, fan-cooled electric motor, it is thermostat controlled. Model V300 may be used with heaters from 25,000 to 300,000 Btu input.
Reznor Manufacturing Company, Mercer, Pa.

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Do you need 6,7, 8 or 10 fins per inch?

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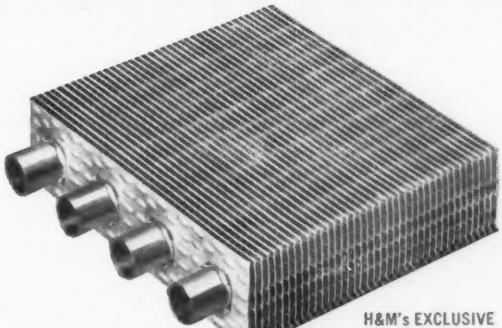
Why accept someone else's "standard" when your own judgment dictates another fin spacing for the specific job at hand. Now you can call on Halstead & Mitchell for Turbu-Flo coils with 6, 7, 8 or 10 fins per inch to meet your individual size requirements.

H&M makes coils for use with most refrigerants, steam (standard or non-freeze coils), hot or chilled water. All coils are available from 1 to 8 rows deep, with finned heights of 12 to 36 inches, and finned lengths up to 10 feet. Aluminum fins on seamless copper tubing are standard, though copper fins may be ordered. Casings come in heavy gauge galvanized steel or aluminum, with connections for left or right hand.

When perfect performance depends on correct design, specify H&M coils. Ask your wholesaler for complete information or write to Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa.



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SOME ANACONDA "FIRSTS" THAT HELP CUT COSTS IN AIR CONDITIONING AND REFRIGERATION

"Tri-Wall Pak" on pallet carrying bunch coils of copper refrigeration tube ⅝" O.D. x .022" gage in lengths 700 to 750 feet long. Note ease of stacking and ready access to coils. Empty cartons are simply folded flat for compact disposal.

Disposable reel carrying 700-foot coil of copper refrigeration tube ⅝" O.D. x .025" gage. Net weight of tube, 124 lb., weight of reel, 12 lb. Disposable reels are furnished in a number of sizes —eliminate returns and bookkeeping on empties.

In the pictures above you see two examples of Anaconda "firsts" that are helping the refrigeration and air-conditioning industry cut costs and improve the efficiency of their equipment.

LONG COILS: Anaconda was the first to provide refrigeration-quality seamless copper tube from .032" to ¾" O.D. and various wall thicknesses in unjointed lengths up to 5,000 feet, depending on size. Coiled tube is furnished in level-wound, bunch, pancake, or semipancake coils to fit individual needs. Level-wound coils can be supplied on disposable reels.

Long coils have many advantages. They provide uniform O.D. and wall thickness for automatic finning . . . save set-up and threading time . . . produce a minimum of scrap ends, and frequently eliminate the need of a standby operator. When changes are made in model design, long coils in stock make change-over in lengths easy and economical.

SPECIAL PACKAGES: Anaconda Copper Refrigeration Tube is shipped in a number of specially designed packages to protect the tube in shipment and handling, to provide compact storage, and to permit high-speed unwinding on automatic continuous-feed fabricating equipment. These include lightweight, sturdy, disposable reels that end returns and bookkeeping on empties. And the "Tri-Wall Pak" makes for compact, safe inventory storage and easy access to individual coils.

Whether your requirements for refrigeration tube are standard or special, Anaconda can provide the size of tube and coil that will satisfy your needs best, packaged to meet your stock room and fabricating requirements most economically. For further information, write: The American Brass Company, Waterbury 20, Conn. In Canada: Anaconda American Brass Ltd., New Toronto, Ontario.

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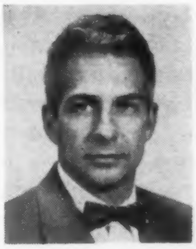
OFFICIAL PUBLICATION ASHRAE JOURNAL
NOVEMBER 1959

Pressurizing High temperature water systems

Preventing the formation of steam by increasing within-the-system pressures, though temperatures rise above 212 F, high temperature water systems offer long appreciated operating and economy advantages. How to pressurize thus becomes the problem and there are several ways in which desired results may be obtained, as authors Ziel and Blossom here summarize.



J. S. BLOSSOM
Member ASHRAE

P. H. ZIEL
Member ASHRAE

Pressurization, above atmospheric, in high temperature water (HTW) systems prevents the formation of steam as the temperature of the water is raised above 212 F.

Development of the various methods of pressurization now in use has evolved out of past practices of using air in conventional hot water heating systems or the use of below-the-water-line hot water from steam boilers for space heating. The use of air develops into various mechanical gas pressurization methods, while using boiler water for heating evolves into the steam pressurization cycles.

Further, the desire to use specific types of boilers will of necessity lead to specific methods of system pressurization. Definite cost data which evaluate the more common methods of pressurization are not generally available, and facts relating to one such case are presented here.

DESIGN CRITERIA FOR PRESSURIZATION

Criteria to be considered when choosing a satisfactory means of maintaining the pressure in the system are essentially as follows:

P. H. Ziel and J. S. Blossom are Consulting Engineers, Ziel-Blossom and Associates.

Minimize fluctuations in the system pressures, temperatures, flow rates and heat content — An attempt should be made to minimize these characteristics if the inherently stable conditions of high temperature water are to be used to advantage. As the fluctuation of these elements is reduced, the following gains may be experienced:

Combustion efficiencies will be improved by virtue of the steady and continuous firing rates.

The variation in water volume of the system due to thermal expansion will be reduced, thus permitting the use of smaller compression tanks.

Upsets in controlling instruments will be avoided as flow rates become steady.

The initial balance of the system will be sustained.

Detect average heating loads and translate them into the control of firing rates at the generator — In furthering stability of the high temperature water system, it is desirable to obtain a means of measuring the average instantaneous load demands of the system. If this can be accomplished, it will permit continuous firing rates of average magnitude rather than excessive peaks or shutdowns. A high temperature water system that depends on the detection of the load by a change in either return or supply temperature at the boiler house is handicapped by a delay until the lower (or higher) temperature water travels from the point of load to the point of measurement. In systems covering large areas, the delays may be from 30 min to 1½ hr. During this time, the boilers cannot supply heat at a rate proportionate to the average system load. It is, therefore, advantageous to be able to detect and answer loads at the boiler house instantaneously, no matter where in the system they occur, especially if the loads are large.

Maintain system pressure to prevent formation of steam regardless of the change in temperatures — A means must be provided to impose a pressure upon the system that

will prevent the formation of steam regardless of the variation of temperature in any part. It is further desirable that this pressure be automatically varied in a direct relationship to the system loads. During periods of maximum (design) loading and the lowest return water temperature, the system pressure must be maintained at a level that prevents the formation of steam in the highest and furtherest reaches of the piping network. The minimum pressure at any point must be in excess of the saturation pressure corresponding to the temperature at that point. As the load decreases with its accompanying rise in return water temperature, the system pressure should be increased to maintain the required departure from saturated liquid conditions. Should some malfunction of equipment cause the formation of steam in any part of the system, then the pressure should be increased even further than its normal limits. At the same time the burner input should be reduced. If feasible, both reactions should be simple and automatic.

Simplicity – The number of pieces of apparatus and their complexity should be kept to a minimum. Construction of equipment should be simple but durable. The controlling devices must be few, but of a nature that requires little attention. These devices should also be of a type that can be serviced by normally available operators.

Minimize maintenance – Equipment should be of a type that is readily accessible, subject to minimum wear and free from any form of rapid deterioration.

Freedom from corrosion – All parts should be free from corrosion internally or externally. Oxygen release in the system should be avoided due to increased tendency toward corrosion at high temperatures. Because of this factor water make-up in any part of the pressurizing elements should be kept to a minimum.

Low costs – First costs and operating expenses must be in line with any advantages in operation if the design is to be economically sound.

Flexibility in locating the compression tank – The design of the pressurizing elements must be such that the compression tank can be placed in the system wherever required to give the best system performance. The compression (or expansion) tank constitutes the one place in the system that is a point of constant pressure whether the pumps are running or stopped.

METHODS OF PRESSURIZATION

Several methods have been suggested for pressurizing. The following paragraphs describe some of these methods and their ability to fulfill the eight criteria previously discussed.

1 – Elevated Tank – An elevated storage tank is the simplest means of pressurizing. This is usually not practical because of the great height required for the pressures encountered. A system of this type does not have the ability to vary pressures other than predicated by the height of the water column. The first six criteria described are reasonably fulfilled. The primary objection to this method is the cost and availability of suitable terrain to run the piping and elevated tank.

2 – Hydraulic Pump – A hydraulic pump may be used to impose an artificial head on the system. Without a cushion tank, however, the large pressure changes encountered in this system with small changes in volume make this highly impractical. Furthermore, it fulfills none of the functions described.

3 – Saturated Steam Cushion – The most common means of pressurizing is with a steam cushion in an upper drum of a boiler or a separate accumulator. In a system using a standard steam boiler, the water is obtained from the steam drum by the addition of a dip tube that extends down through the steam space into the liquid. In a system using separate cushion tank, the water is drawn off the bottom of the tank. Since the water in either case is at the vaporization point (saturated liquid), it cannot be handled directly without cavitation or flashing in the pump suction, unless some means is provided to increase the net positive suction head. Elevating the cushion tank to produce a high static head may alone serve this purpose. In this type system it is necessary that the pumps be placed so that they draw water from the boiler or cushion tank. Were the pumps located to discharge into the cushion tank, the temperature - pressure balance of the saturated liquid would be upset. An upset in this balance would cause flashing, not only in the cushion tank, but in various parts of the system. Any variation in system flow due to the throttling action of control valves or the shutdown of any one pump would cause sufficient upset to produce flashing.

To prevent the preceding condition, a bypass is provided from the return line to the pump suction connection. The bypass makes it possible to mix saturated liquid

Fig. 1 High temperature water elevated tank pressurized (method 1)

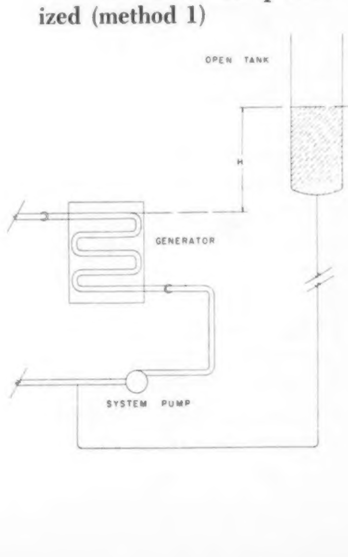
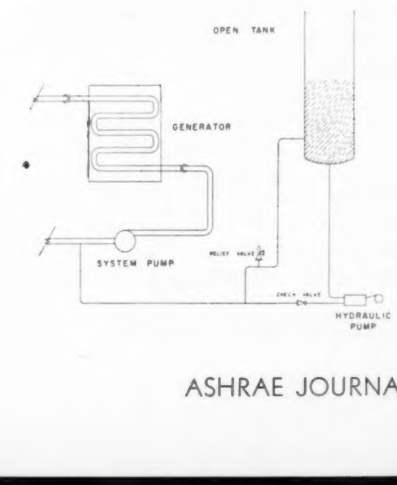


Fig. 2 High temperature water hydraulic pump pressurized (method 2)



from the cushion tank with cool return water. The mixture is cooler than the saturated liquid, but is still at saturated pressure. The mixed liquid temperature must be sufficiently below the boiling point to overcome cavitation. This degree of mixing is critical and must, therefore, be carefully controlled. During periods of low load it is possible to experience return water temperatures so high that no amount of mixing will give the necessary departure from saturated liquid.

This system fails to accomplish the eight design criteria as follows:

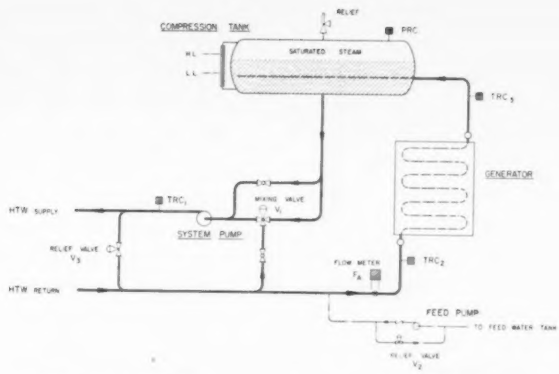


Fig. 3 High temperature water cycle saturated steam cushion, one pump system (method 3A)

Minimize fluctuations

On a one pump system (3A), bypasses from the return piping into the system circulating pump suction result in reduced flow rates through the boilers. Reduced flow rates through the boilers due to bypassing are likely to cause vapor locking in the boiler tubes and subsequent burning. Boiler circulating pumps are necessary to avoid this condition.

As the load decreases in a one pump system, the temperature will rise above the control point since bypassing into the circulating pumps is the only means of departure from the saturated-liquid temperature.

A two pump system (3B) will suffer the same disadvantage if the system pumps take water from the cushion tank.

Steam formation in the boiler or cushion tank, caused by flashing due to reduced pressure, will increase further during periods when the system pumps are handling greater quantities of water than that circulated through the hot water generators. The excess quantity of water returned at lower temperature directly to the cushion tank will cause partial condensation of the pressure cushion. To maintain the pressure, the boiler has to flash off additional steam and could under extreme conditions cause vapor locking in boiler tube circuits with certain designs.

Detection of average loads is not possible since changes in the bypass control affect the system volume-temperature relationship.

Maintain pressure differential above saturation

Temperature and pressure are interdependent with no separation possible.

Boilers must be brought on the line with extreme care lest the slight departure from the compression tank pressures cause severe water hammer.

Circulating pumps always operate under the worst (lowest) net positive suction head of any point in the system.

Simplicity

Compression tanks are complex due to the necessity of internal distributors, mixing devices, blow-off connections, and the many reinforced nozzles required to serve them.

Water level must be maintained more closely for proper steaming, mixing and distribution.

Air must be vented from the compression tank before steam can occur if it stands high in the system. If the tanks are low, thus producing a flooded compression tank before firing is begun, the system may be manually valved off, the tank drained, and steam produced before the system circulation can be achieved. Whenever firing is stopped, the tank will produce a vacuum unless relieved by allowing air to return into the tank. Air is a contaminant not to be allowed in the system, if possible.

Maintenance of control bypasses, cushion tank venting, numerous hand valves plus the difficulty of bringing cold boilers on the line results in a maximum effort to keep the system tight and in efficient operating condition.

Corrosion

The compression tank can suffer internal corrosion from sludge deposited by the flow of system water through the tank. Further damage may result from oxygen released in the heated water or carried back in the return lines.

Since the tank is usually opened to atmosphere to relieve a vacuum when the system is shut down, oxidation is unavoidable.

First costs of the cushion tank are relatively high due to the need for heavy insulation to minimize radiation heat losses, structural steel tank supports, internal distributors and numerous items of trim, and large nozzle connections due to full system flow through tank.

Flexibility in selecting the location of the cushion tank is almost nonexistent. The tank must be set above and near the boilers if it is to function as a steam accumulator.

4—Mechanical Steam Cushion—A second means of pressurizing with steam is with the use of a separate

steam generating source to provide a cushion in the compression tank. As in the preceding paragraph, this compression tank could carry the full flow from the boilers. However, the water flowing would not be at saturated conditions. The external steam generator could maintain the cushion pressure as high above saturation as desired. This steam generator would be small and simple since the total head input would be equal only to the radiation losses of the cushion space in the compression tank. A normal 6 ft-0 in. diam by 30 ft-0 in. long tank set vertically would require approximately seven kilowatt heat input to maintain pressure in a tank containing 400 F water and covered with 2 in. of 85% magnesia block. It should be noted that the tank is set vertically to reduce the radiational surface of the cushion space.

The steam generator might consist of an immersion electrical resistance heater controlled from a pressure switch in the compression tank. This system eliminates many of the failings found in the normal saturated steam cushion. The desired criteria fulfilled are as follows:

100% system flow can be maintained through the boilers at all times with only one pump. No manual or automatic bypass valves or controls are needed.

The system will stay at a temperature control point irrespective of pressure or load.

Steam formation or vapor locking of boiler tubes is eliminated.

Circulating pumps can be located on the return side of the system and therefore operate with the greatest available net positive suction head.

Since the means of pressurization is from a separate energy source than the means of heating the water, the system pressure has been made independent of temperature.

Accurate water level control would not be required in this vertical tank and full system expansion could be accommodated without the use of reliefs or hydraulic feed pumps.

A vertical compression tank should be substituted in lieu of the normal horizontal vessel to minimize the radiation surface in the steam cushion. This vertical tank would contain no internal distributors, mixing devices, or large reinforced nozzles. Costs are thereby reduced.

The compression tank can be located at any level or point in the system making it more flexible.

Method No. 4 fails to fulfill certain criteria:

It is not possible to detect and answer loads at the boiler house instantaneously, no matter where in the system they occur. This is true because pressure is controlled as a constant. Pressure could be reset by sensing a change in return water temperature. This sensing, however, must wait until the water supplying any load returns to the boiler room.

The compression tank must be insulated to minimize the heat losses by radiation.

5—Mechanical Gas Cushion—Three basic designs of gas pressurization have been used in various instances as follows:

The first system (5A) uses a compression tank containing a fixed quantity of gas. No system water flows through the tank. The tank floats on the return main by means of a small interconnecting balance line. The balance line is so piped to avoid thermal convection and heating of the tank. Gas pressure is applied to the cushion space in an amount predicated by the system design. This pressure must correspond to the maximum static liquid head on the system, plus a margin of safety above the saturated liquid point.

With the system flooded any variation in water volume will change the water level in the compression tank. As the water in the system expands, it will cause a rise in level in the tank. Upon reaching a predetermined level, a control valve will automatically be opened allowing water to flow into a large low-pressure receiver. The valve likewise closes when a lower limit is reached.

As the water level falls within the compression tank due to system contraction, a feed pump is started which returns water from the low pressure receiver into the system. A high limit switch likewise stops the pump.

Normally the compression tank can be relatively small since the low pressure receiver is sized to hold most of the water of expansion.

Fig. 4 High temperature water cycle saturated steam cushion, two pump system (method 3B)

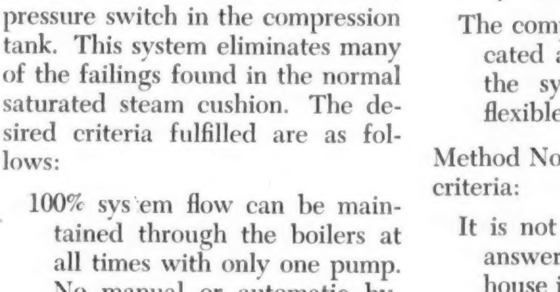
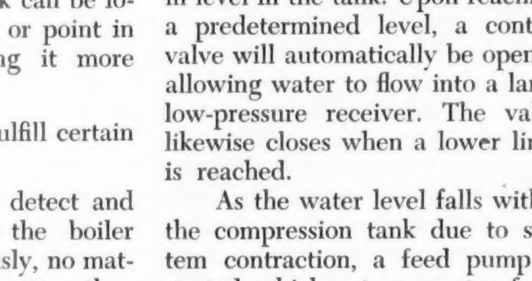


Fig. 5 High temperature water cycle saturated steam cushion, one pump system (method 4)



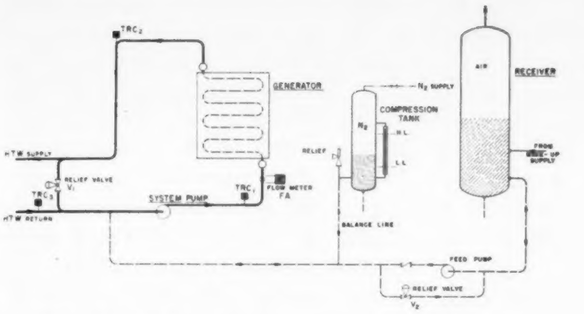


Fig. 6 High temperature water cycle mechanical gas cushion (method 5A)

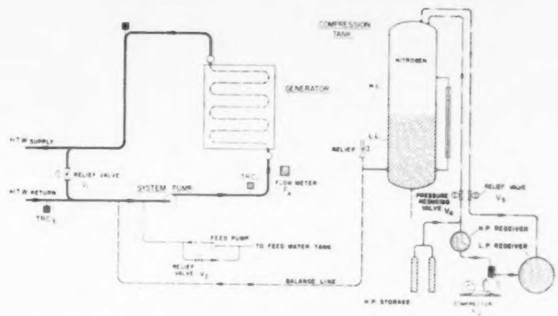


Fig. 7 High temperature water cycle mechanical gas cushion (method 5B)

This system has advantages over the saturated and mechanical steam cushion as follows:

100% system flow can be maintained through the boilers at all times with only one pump, essentially the same as Method 4.

The margin of safety below the saturation curve can be increased or decreased at any time, independent of operating temperature.

Accurate water level control is not required in the compression tank since level serves no function in this system.

No venting or vacuum relief is required in the compression tank. The gas charge is supplied only once at the initial operation of the system.

A vertical compression tank and receiver can be used, each having only a few small couplings for connections. No internal fittings are required. Tanks can be skirt mounted, thus requiring no external support.

The compression tank or low pressure receiver does not require insulation since each floats on the line cold.

Disadvantages are:

A common failing with the preceding system is that it is not possible to detect and answer loads at the boiler house instantaneously regardless of where they occur in the system.

A disadvantage peculiar to this system is in the two pressure vessels required (the compression tank and receiver), plus the continuous use of a relief valve and feed pump. Failure in either of these elements will cause malfunction of the system.

Corrosion from oxygen picked up in the low pressure receiver remains a problem as in all the preceding systems.

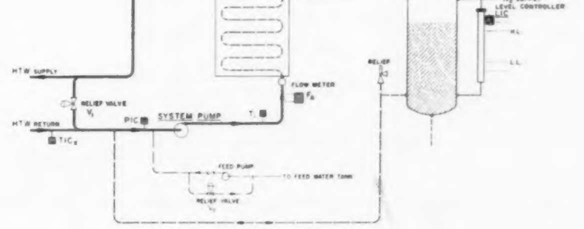
A second gas pressurized system (5B) is one in which the compression tank is sufficiently large to contain the total water of expansion. In this respect, it is identical to the mechanical steam cushion. As in the mechanical steam cushion system, the mechanical pressurization is maintained at a constant point by the action of an external apparatus. In this system, the compressed gas in the cushion part of the cylinder is maintained at a predetermined pressure. As the liquid level rises because of system expansion, the compressed gas is relieved through a control valve into a low pressure cylinder. When the pressure has been reduced to a low limit, the control valve will close. Gas in the low pressure receiver is pumped by means of a small compressor into a high pressure receiver. As the liquid level in the compression tank decreases, with increased system load, a control valve will open to admit gas from the high pressure receiver. Upon reaching a high pressure limit, the control valve will close. This system has the same advantages and disadvantages of the preceding gas pressurized system. Further, it fails in the detection to answer loads at the boiler house.

The third method of gas pressurization (5C) is with the use of a compression tank sized to take the entire system expansion and containing a fixed weight of gas. As in the other gas pressurized systems, the compression tank floats on the system return line by means of a small balance pipe connection. It should be noted that all of the preceding methods of pressurizing failed to fulfill at least two of the functions set up as desirable.

System 5C does not contain gas compressors, feed pumps, or low pressure receivers. The compression tank contains a fixed charge of gas that is never replaced or varied. It cannot be lost since the cushion space in the upper part of the tank has no openings.

This system is based on the use of the compression tank as a thermometer to indicate with acceptable accuracy the mean temperature of the volume of water

Fig. 8 High temperature water cycle mechanical gas cushion (method 5C)



contained in the piping and apparatus. Assuming that the constant mass of water in the system is incompressible, its volume is a function of mean temperature only. If a change in volume can appear only in the compression tank, and if the compression tank level is a true measure of volume, then measurement of this level will be a true measure of mean system temperature.

True, a change in mean system temperature will be caused by a change in system load, which will appear as a change in compression tank level instantaneously, assuming water to be incompressible. The use of this change in level to reset the boiler outlet thermostat, which in turn controls the burner combustion rate, constitutes an immediate response to load.

Placing a cylindrical compression tank with its axis in a vertical position, a change in level becomes a linear measure of change in volume. The fixed mass of gas results in the maximum protection against steam formation with minimum pressure in the compression tank. At the lowest mean system temperature, and correspondingly low return water temperature, the compression tank pressure will be at its lowest value with the system water most capable of absorbing full burner output. At higher system temperatures, the compression tank pressure will rise while the burner output is decreased, giving greatly increased safety at higher return water temperature.

Thus the gas pressure in the tank varies automatically through a range determined by the tank volume in relation to system volume, and the range of water temperatures as determined by the control system.

It is therefore seen that the gas pressurized system just described (5C) fulfills all of the desired functions of operation. Not only does it fulfill the two functions not found in any of the other systems, but fewer pieces of apparatus and controls are required, the compression tank is the least complicated, and therefore, the lowest in cost. No insulation is required on the compression tank since it floats on the line cold. The central heating plant consists of the system circulating pumps, the boilers, and a single compression tank. If future expansion is indicated an additional compression tank can be added without causing complications in the operation.

SELECTION OF GAS FOR PRESSURIZATION

Consideration should be given to the characteristics of the gases used for pressurization in respect to corrosion. Nitrogen has been used as well as compressed air in several different types of applications.

A compression tank using compressed air as the pressurizing media may suffer extreme corrosion because of the alternate wetting and dry conditions at the water line. This water line corrosion could be limited, however, if only a fixed quantity of air were used. The small amount of oxygen contained in the air would soon oxidize sufficient metal so that no more would be available to continue the reaction.

Nitrogen might, however, be a better selection since the cost of an inert gas is a small part of the system. Nitrogen is the least expensive and most widely distributed relatively inert gas commercially available for this purpose.

To further preclude the possibility of corrosion, the compression tank should be so piped to the return line that entrained gases boiled off from the initial fill of water be kept from entering the tank.

COMPRESSION TANK SIZING

The many variables affecting the size of the compression tank make the selection of tank size based solely on system volume and temperature change in the water a relatively uneconomical procedure. The method of pressurization and of system control are basic factors affecting the possible total change in system volume to be accommodated in the compression tank.

One attempt to show the vari-

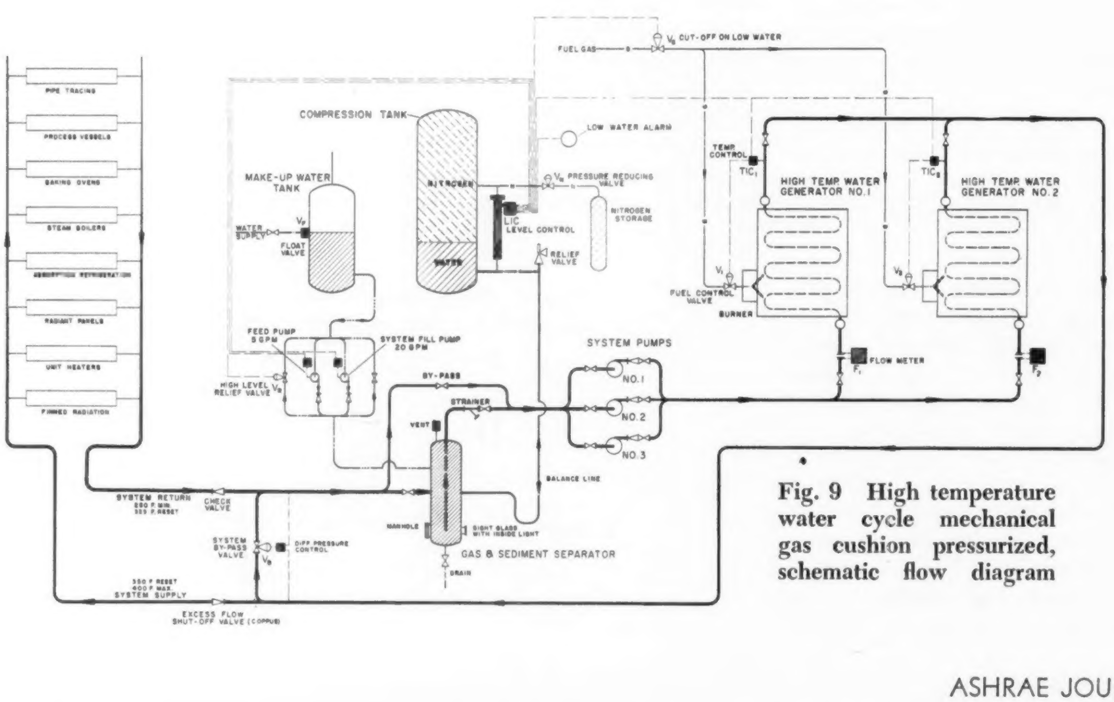


Fig. 9 High temperature water cycle mechanical gas cushion pressurized, schematic flow diagram

ation possible in design is presented in Fig. 9, which shows an HTW system mechanically gas pressurized by method No. 5C. The compression tank accommodates the total system change in water volume. Variations in tank size for this specific system when operating at a maximum temperature of 400 F with various types of system control have been analyzed. Tank size varies from 125% to 10% of system volume. This is a range of 12.5 to 1 or 1,250%. In this case, the pressure change in the tank has been limited by design from 300 psig to 350 psig. If other values are used for the pressure limits in the system, further large variations in tank size will be determined.

While the sizing of the tank is based on fundamental physics, the effect of system design and control is an integral part of the sizing problem. In larger systems, it should be of primary concern in tank selection. In general our tank size is not finalized in the system design until the control scheme is determined. Five specific examples of tank sizing are given in the following section.

System Number	Type System	Pressuriz. Method	Data Source for Sizing Compression Tank
V	Saturated steam cushion, one pump cycle	3A	Corps of Engineers System
W	Saturated steam cushion, two pump cycle	3B	Corps of Engineers System (Full temperature range)
X	Saturated steam cushion, two pump cycle	3B	Ziel-Blossom (Reset temperature range)
Y	Nitrogen-gas cushion, one pump cycle	5C	Ziel-Blossom (Full temperature range)
Z	Nitrogen-gas cushion, one pump cycle	5C	Ziel-Blossom (Reset temperature range)

COST STUDY OF TWO METHODS OF PRESSURIZATION

The following study was made to determine the installed costs of the major equipment used in the central heating plants of several HTW systems. Since the system design is an integral part of the method of pressurization, it was believed that a more realistic cost comparison would result by using total central plant system costs.

Three steam-cushion designs identified as V, W, X, and two nitrogen-cushion designs identified as Y, Z have been compared.

A. Design Conditions — Operating conditions are the same for all five systems. The following data were assumed for design purposes:

Total system load—50,000,000 Btu/h.
System pressure drops for pumping.

Distribution piping—dynamic head —100 ft water.

Distribution piping—static head—30 ft water.

Central plant piping—dynamic head —one pump cycles—25 ft water.

Central plant piping—dynamic head —two pump cycles—40 ft water.

HTW flow in system—6,450 lb per min.

HTW supply temperature—400 F.

HTW return temperature—250 F.

Water volume in system at 60 F—14,000 gal (1,872 cu ft)

Fuel for HTW generators—natural gas with No. 6 oil standby.

B. Major Equipment Selection—

HTW Generators. Water tube LaMont type with combination gas oil burners, combustion controls and trim. Two units each at 60% of the total load or 30,000,000 Btu/h.

Make-up Water Treatment and Pumps. Complete with associated trim items and based on an estimated capacity of 120 gal per day for either system.

HTW Pumps. Refinery type with mechanical seals.

One pump cycles (Systems V, Y, Z) use three system circulating pumps, one is a standby, each sized to match each HTW generator for 471 gpm as 125 ft head with 30 hp drivers.

Two pump cycles (Systems W, X) use three generator circulating pumps, one is a standby, each sized for 471 gpm at 40 ft head with 15 hp drivers; and two system circulating pumps, one is a standby, each sized for 785 gpm at 100 ft head with 30 hp drivers.

Compression Tanks

For system V the tank has been sized by using the Corps of Engineers 1953 Manual Procedure as follows:

1,872 cu ft water at 60 F at 0.01604 cu ft per lb = 116,700 lb

116,700 lb at 212 F at 0.01672 ft^3/lb = 1,951 ft^3

116,700 lb at 360 F at 0.01811 ft^3/lb = 2,113 ft^3

(360 F = maximum mean temperature)

116,700 lb at 415 F at 0.01886 ft^3/lb = 2,201 ft^3

(415 F = maximum temperature)

Volume increase in system,
from 212 F to 415 F = 2201 — 1951 = 250 ft^3
from 360 F to 415 F = 2201 — 2113 = —88

Net volume increase = 162 ft^3

The maximum tank level change for the 162 ft^3 is limited to 18 in. Assuming vertical tank sides the tank area at the center line is 162 ft^3 divided by 1.5 ft = 108 sq ft.

With a 5 ft-10 in. I. D. tank, the required length is 108 sq ft divided by 5.833 ft = 18.5 ft long (or 18 ft-6 in. shell length).

For system W the tank has been sized by using Corps of Engineers 1953 Manual Procedure as follows:

Volume of system water at 60 F = 7,872 cu ft.

Volume of 116,700 lb system water at 400 F at 0.01864 ft^3/lb = 2,175 ft^3

Volume of expansion for water = 303 cu ft. Note: the Corps criteria require the tank to be sized for the full range of temperature change in the water.

The example given for tank size allows 45% of total tank volume for required water expansion. Tank size required is 303 divided by 0.45 = 673 cu ft.

673 cu ft × 7.481 gal per cu ft = 5,035 gal compression tank capacity.

We have selected a tank 5 ft-10 in. inside diam and 24 ft-0 in. long between dished heads for this capacity.

This method includes in the 55% of remaining tank space the allowance for expansion of the water required in the tank to establish the minimum level.

For system X the tank has been sized as follows:

Water Conditions for the distribution system.

Full Load

400 F maximum leaving temperature.

250 F minimum return temperature.

325 F mean temperature at maximum load.

Low Load

360 F minimum reset temperature.

340 F maximum return temperature.

350 F mean temperature at minimum load to be used as maximum system temperature since it exceeds mean temperature at full load.

Weight of water = 80% of total (assumed) = 0.8 × 116,700 lb. = 93,360 lb.

Water Conditions for the Boiler Distribution Circuit.

409 F maximum operating temperature.

60 F minimum temperature—fill.

Weight of water = 20% of total =0.2 × 116,700 lb = 23,340 lb.

Compression Tank Characteristics as follows:

50% steam space

10% water volume and tank water expansion

40% water expansion of system.

Total System Expansion

Boiler Circuit (23,340 lb) (.01877 ft^3/lb at 409 F) = 438.1 ft^3

less (23,340 lb) (.01604 ft^3/lb at 60 F) = 374.4 ft^3 = 63.7 ft^3

System Expansion (93,360 lb) (.01799 ft^3/lb at 350 F) = 1679.5 ft^3

less (93,360 lb) (.01604 ft^3/lb at 60 F) = 1497.5 ft^3 = 182 ft^3

Total System Expansion = 245.7 ft^3

Tank Size if 40% used for 245.7 ft^3 expansion:

$$\frac{245.7}{40} = 614.3 \text{ cu ft}$$

614.3 ft^3 × 7.481 = 4,596 gal.

A tank with 5 ft-10 in. inside diam will have a shell length of 21 ft-9 in. for this volume.

For system Y the tank has been sized as follows:

Volume of water expansion (10% water volume and tank water expansion) = 303 cu ft.

Minimum system pressure at 60 F — assumed = 100 psia.

Maximum system pressure at pump discharge = 315 psia.

Less system pumping head of 125 ft or 54 psi = 261 psia maximum pressure at compression tank.

Determination of minimum gas volume (which is at maximum water volume) from Boyle's law. Since the tank operates cold and at a relatively constant temperature, we have ignored temperature effects.

P_1 (min. tank pressure) = 100 psia.

$V_1 = V_2 + 303$ cu ft water expansion.

P_2 (Max. tank pressure) = 261 psia.

V_2 = minimum gas volume.

Therefore: $(100)\ (V_2 + 303) = (261)\ (V_2)$

$V_2 = 188$ cu ft

$V_1 = 188 + 303 = 491$ cu ft or 3.673 gal.

Allow approximately 10% or 400 gal for minimum water in tank plus 100 gal for sludge.

Total tank size equals 4,173 gal or 558 cu ft.

A tank with 5 ft-10 in. inside diam will have a shell length of 19 ft-8 in. for this capacity.

For system Z the tank has been sized on the basis of resetting the maximum water temperature to 360 F and reducing the minimum system pressure to 65 psia as follows:

Volume of 116,700 lb at 360 F at at 0.01811 ft^3/lb = 2,118 ft.

Volume of 116,700 lb at 60 F at 0.01811 ft^3/lb = 1,872 ft.

Net system expansion 241 ft.

Determination of minimum gas volume:

$P_1 = 65$ psia $P_2 = 261$ psia

$V_1 = V_2 + 241$ ft water expansion

V_2 = minimum gas volume

$(65)\ (V_2 + 241) = (261)\ (V_2)$

$V_2 = 80$ cu ft

$V_1 = 80 + 241 = 321$ cu ft or 2,401 gal

Allow 10% or 240 gal minimum water plus 100 gal for sludge.

Total tank size equals 2,742 gal or 367 cu ft.

A tank with 5 ft-10 in. inside diam will have a shell length of 12 ft-6 in. for this capacity.

Table I summarizes the tank characteristics for the five systems for 5 ft-10 in. inside diam at 300 psig design with 1 1/16 in. thick shell. Weights include all nozzles and/or internals as used for each respective system but do not include saddles, rollers or supporting steel.

SYSTEM COSTS

Those costs used herein for all equipment are based on current quotations from major manufacturers of the specific products concerned. Equipment costs include associated trim and control items. Pipe fittings, insulation, and labor were estimated by actual material and labor take-offs from preliminary design sketches and priced at the current market. All costs are as if contracted-in-place and include margins for contractors' overhead and profit.

The compression tank prices include as they apply for each system the costs of the tank; associated openings, internals and trim; and insulation.

Item 7 for piping and labor includes all pipe, valves, fittings, piping accessories, pipe insulation and labor for piping, rigging and equipment setting. The piping system is based upon 300 psi design.

Not included in these costs are the separate building for the mechanical equipment, any HTW piping or equipment outside the central plant, HTW generator breechings, fuel oil storage, natural gas service or piping, electrical power service or distribution wiring or concrete foundations.

COST EVALUATION

Note that the basic cost differences stem from the simplicity of the mechanical gas cushion system as compared to the saturated steam cushion system. Sizes of compression tanks will vary from these selections depending upon temperature and pressure ranges selected in the computations.

The cost of items 2 to 7 in Table II for the saturated steam cushion systems as against the same items in the mechanical gas cushion systems show differences ranging from $34,240 to $14,670 in favor of the mechanical gas cushion.

The fact that the mechanical gas pressurization method 5C best fulfills the design criteria and has the lower first cost as shown in systems Y and Z warrants primary consideration of this method in the design of HTW systems in all size ranges.

TABLE I

No.	Gal	Length	Weight	Surface
V	3,949	18 ft-6 in.	21,144 lb	424 sq ft
W	5,049	24 ft-0 in.	26,216 lb	528 sq ft
X	4,599	21 ft-9 in.	24,375 lb	485 sq ft
Y	4,173	19 ft-8 in.	19,500 lb	446 sq ft
Z	2,750	12 ft-6 in.	13,633 lb	310 sq ft

TABLE II SYSTEM COSTS

Item of Cost	V Steam	W Steam	X Steam	Y Nitrogen	Z Nitrogen
1. HTW generators, burners, controls	$127,570	$127,570	$127,570	$127,570	$127,570
2. Water treatment equipment	1,220	1,220	1,220	1,220	1,220
3. HTW pumps	10,380	15,730	15,730	10,380	10,380
4. Compression tank	12,980	15,460	14,660	7,850	5,720
5. HTW temperature controllers	4,060	4,060	4,060	1,350	1,350
6. Nitrogen equipment	None	None	None	580	580
7. Piping & labor	36,470	46,080	46,080	29,060	29,060
Items 2 to 7 Sub-total	65,110	82,550	81,750	50,440	48,310
Total System Cost	$192,680	$210,120	$209,320	$178,010	$175,880

Free-piston compressors

may power air conditioning systems

J. H. McNINCH

R. J. McCRORY

R. W. KING

Answering a question posed by the American Gas Association, "How can the potential energy in natural gas be transformed into a refrigerating effect with high efficiency and low first equipment cost in such a way as to be attractive for residential air conditioning service?", the Battelle research group developed the free-piston compressor. The approach was to combine the spark-ignition gas engine and the vapor-compression refrigeration cycle in such a way that the initial installment, maintenance costs, and the noise level would not be deterrents to its general acceptance. The compressor is designed for efficiency and low cost production, but it does not yet have a manufacturer.

Conventional internal combustion gas engines have many advantages, but first cost, as well as operating and maintenance costs, and noise level could be deterrents to general acceptance as an air-conditioning power source. The free-piston engine compressor, however, does combine power generation and vapor compression functions in a single unit in a way which promises to overcome the primary objections cited above. Also, the resulting combination is compatible with the large number of existing air conditioning systems employing commercial refrigerants, and could be installed as a remote unit for new or replacement systems.

Predominant among features of the free-piston refrigerant compressor is its unusual simplicity. With the exception of pressure-actuated valves which have slight motion, the only moving part is the piston. Power developed in the combustion chamber is transmitted directly through the piston to the refrigerant. With few parts and essentially no critical fits, the production cost of the free-piston compressor promises to be low. The inherent simplicity of the free-piston unit will also help to satisfy the requirements for reliability, a primary factor in establishing maintenance costs. Furthermore, extremely conservative design practices can be and were employed as the development proceeded. With essentially a single moving part, pneumatic noise can be limited to the inlet and exhaust of the power end, and can be satisfactorily muffled.

Description – The free-piston refrigerant compressor has a two-stroke-cycle combustion chamber on the top end and a compressor cylinder with conventional inlet and discharge valves on the bottom end. The diameters of both power and compressor cylinders are 2¾ in., and the nominal stroke of the piston is 3 in., resulting in nominal displacements of 17.8 cu in. The operating frequency is 1500 cpm with a condenser saturated temperature of 110 F. The refrigeration capacity under these conditions is 3 tons.

At the present time, development of the free-piston refrigerant compressor for use in a gas residential air conditioning system has proceeded to a point where technical feasibility has been established. Many of the details have been worked out and several units are operating successfully in the Battelle laboratory under simulated field conditions.

Further development must be done to satisfy the requirements for residential air conditioning service. Although the unit has been designed with life and durability as prime considerations, substantial work in the laboratory and field remains to satisfy the specified 2000-hr service life and 10,000-hr replacement life. Further development will also improve its per-

R. J. McCrory is Chief, R. W. King Assistant Chief, and J. H. McNinch a Project Leader in the Mechanical Research Div of the Battelle Memorial Institute. This paper, here published in condensed form, is based on a program of development sponsored at Battelle by the American Gas Association. It was presented at the ASHRAE annual meeting in Lake Placid, N. Y., June 22-24, 1959 as "Free-Piston Compressor in an Air Conditioning System". The complete paper will appear in ASHRAE Transactions.

formance and decrease its operating cost.

The free-piston refrigerant compressor can be adapted for other applications and other fuels, including LPG, gasoline, and fuel oil, on residential heat pumps, automotive air conditioners, and truck and railroad car refrigerators. After specific development, the basic advantages of the free-piston compressor can be expected to provide attractive units for these applications.

Operating characteristics – Fig. 1 schematically shows the general design of the compressor. This paper concentrates on the refrigerant compressor end of the cylinder, but presents a brief summary of the operation of the entire unit. The combustion chamber employs loop scavenging with air inlet and exhaust ports in the cylinder walls. Air for scavenging and combustion enters the counter chamber through reed valves where it is compressed and delivered to the scavenge air box. When the piston uncovers the air inlet ports on its down stroke, the air scavenges the spent products of combustion from the previous cycle through the exhaust ports, and a new air charge is entrapped in the combustion chamber prior to the piston's compression stroke.

As the piston covers the exhaust ports, a charge of natural gas is injected from the counter chamber actuated gas injector and, as the compression stroke nears completion, a full charge of gas and air is intermixed in the combustion chamber. Ignition occurs when a ground electrode on the piston crown passes in close proximity to a high-voltage electrode in the cylinder head, and a spark occurs.

As the combustion pressure drives the piston downward, compression begins in the refrigerant compressor and continues until the discharge valve opens and refrigerant flows to the condenser. When compression and displacement of the refrigerant dissipate all of the kinetic energy of the piston, the piston stops its downward travel with the compressor clearance volume filled with refrigerant at condenser pressure. The piston is started on its return by the reexpansion of the refrigerant in the

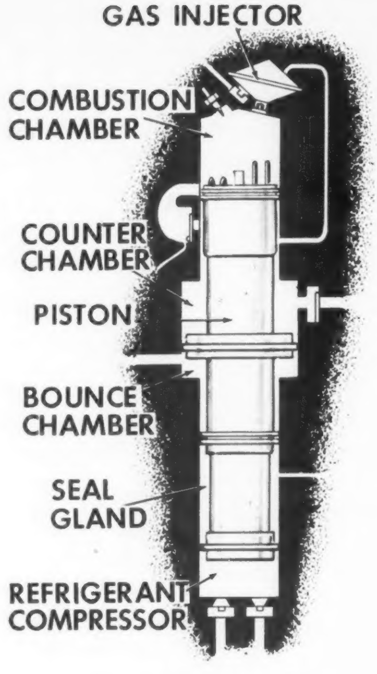


Fig. 1 Free-piston refrigerant compressor

clearance volume and, after inlet pressure is reached in the refrigerant compressor, refrigerant at evaporator pressure provides the remaining energy for the upward stroke of the piston against engine compression pressure.

Operation of the free-piston compressor is obtained over a wide range of operating conditions by careful balancing of the compression and expansion energies in the combustion chamber and compres sor. It was found that the refrigerant compressor automatically adjusts its compression ratio as the pressure ratio changes, so that the work of compression during any one cycle is constant in the range of required pressure ratios.

This means that there is no need to modulate the amount of power generated in each cycle, and that a full charge of air and fuel can be used regardless of compressor conditions. This inherent characteristic of constant compressor work reduced the energy balance problem to one of absorbing the total expansion power of the combustion chamber in refrigerant compression while providing sufficient refrigerant energy to make possible rebound for the piston's upward stroke.

Starting of the compressor is accomplished on the present models by partially evacuating the refrigerant compressor cylinder with a small auxiliary refrigerant compressor. With the piston at a full upward position during shutdown, evacuation causes the piston to move to the bottom of its stroke, and rapid opening of the compressor inlet valve admits refrigerant at shutdown pressure to actuate the piston on its starting stroke. The functions of fuel injection and ignition are identical to those during normal operation, and starting is extremely dependable. However, if the engine should not start, the starting cycle automatically repeats.

Silencing of the inlet and discharge to the combustion chamber is accomplished by standard mufflers having sufficient volume to satisfy the silencing requirements.

The unbalanced force introduced by the reciprocation of a single piston is isolated by a simple spring suspension system which permits the inertia of the cylinder to absorb almost completely the unbalanced forces.

Performance—Fig. 2 shows the capacities of the free-piston unit over a full range of condenser and evaporator pressures. Stable operation of the unit occurs for every point within the bounds of the heavy line. Also shown on this curve are the capacities that are obtained for the various evaporator and condenser pressures.

Fig. 3 shows the "performance factor" for a constant evaporator temperature of 40 F over a range of condenser temperatures. The term "performance factor" is defined as the cooling effect divided by the heating value of the fuel delivered to the unit.

Fig. 4 is a nomograph from which the implications of this high performance factor as it relates to energy cost can be computed for specific natural gas and electric power rates. Although energy costs vary widely according to geographic location and level of utilization, the gas and electric rates for specific situations can be applied to Fig. 4 to obtain the relative energy costs. The example shown is for Columbus, Ohio, where for typical energy rates, the energy cost of the gas free-piston unit would be 55% of the energy cost of the electric air conditioner. Where the energy

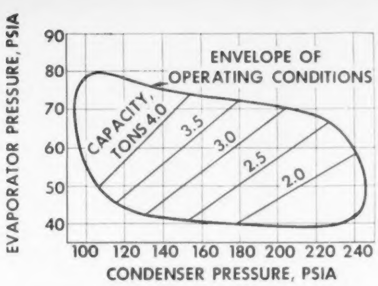


Fig. 2 Operating characteristics of the free-piston refrigerant compressor

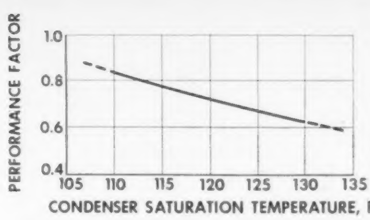
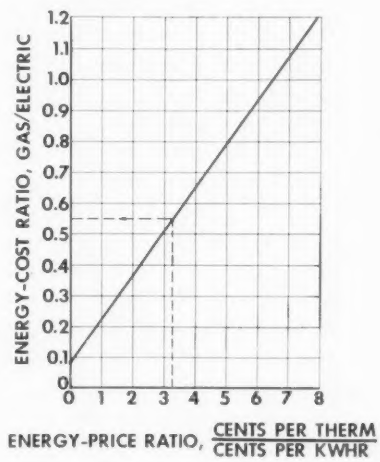


Fig. 3 Performance of the free-piston refrigerant compressor

Fig. 4 Comparative energy costs of 3-ton air conditioning units gas-fueled free-piston unit vs electric-motor-driven unit



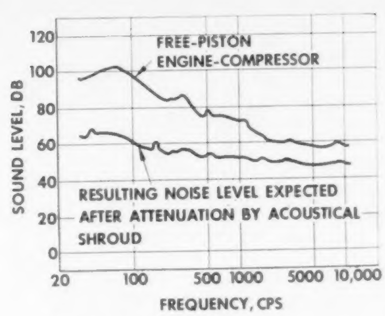


Fig. 5 Noise spectrum of the free-piston compressor

cost ratio is less than 6.5 (cents per therm/cents per kwhr), the energy costs of the gas free-piston air conditioner can be expected to be less than for the electric air conditioner.

Illustrated are assumed energy requirements for gas unit: 0.424 therm per hr; 0.25 kw for cooling. Electric unit: 3.0 kw.

Fig. 5 shows the noise spectrum of the present unit. The higher line is without shrouding of any kind, and the lower line is the anticipated noise level with a cover of the type that would be used for a remote installation. Continuing work is further substantially reducing this noise level.

TECHNICAL DETAILS

Sealing of the refrigerant within the compressor end of the unit is accomplished by maintaining an intermediate fluid in a gland around the compressor piston. By maintaining a sealing fluid such as refrigerant oil, at a pressure in the gland higher than in the refrigerant compressor, any leakage paths past the piston rod are sealed by oil flow from the gland into the compressor cylinder. The sealing fluid is prevented from escaping by hydraulic sealing rings.

Fig. 6 shows a seal design devised to combine the objectives of low viscous friction, low wetted area on the piston skirt, and oil stripping. The principle of operation of the revised seal design was the basic reverse leakage process. As shown in Fig. 6, a ¾-in.-diam piston rod which connects the combustion and compressor pistons decreases both the viscous shear area and the wetted area extending above the hydraulic ring. The volume above the compressor piston (termed the secondary compression chamber) provides a means for obtaining a low pressure for stripping the refrigerant from the sealing oil.

As shown in Fig. 6, a check valve between the secondary chamber and the compressor inlet line causes the maximum pressure in the secondary to be compressor inlet pressure. This occurs at the top of the stroke of the piston, and the secondary pressure decreases as the piston moves down. A check valve between the secondary chamber and a stripper causes minimum secondary pressure to be applied above the oil level in a low-pressure stripper stage. With this design, the oil passes from the separator into a first stripper stage where it is heated, still at condenser pressure. Expansion through a float valve provides a second stripping effect with the stripped refrigerant vented into the secondary compression chamber. The stripped sealing oil then passes into the sealing gland where its pressure is sufficient to seal the refrigerant in the secondary compression chamber.

Fig. 7 shows the refrigerant loss that resulted with this seal configuration with a paraffin-base refrigerant oil, Refrigerant-12, and a stripping temperature of 250 F. As can be seen, low rates of refrigerant loss are obtained by expanding the secondary compression chamber to low pressures.

A large amount of operating experience was obtained with the seal design shown in Fig. 6. However, during the continuous simplification program, the external stripper and associated plumbing were eliminated by a new seal configuration.

Fig. 8 is a schematic drawing of this seal. As shown in the figure, the stroke length sealing oil gland is replaced by a much shorter gland which is supplied with oil directly from the separator. Located below the oil gland is a second gland containing refrigerant which is maintained at a low pressure. With low oil flow rates, it is this low-pressure refrigerant chamber that determines the refrigerant concentration in the exposed oil film; therefore, no external stripper is required. Viscous friction and heat transfer from the piston maintain the oil film at a temperature which results in a low concentration of refrigerant. The performance of this seal configuration is essentially the same as that obtained with the stroke length seal but without the complication of the external stripper.

Contracting metallic oil sealing rings have been developed which dependably maintain oil leakage rates below 0.05 lb per 1000 hr when the unit is operating. Although the rings also show promising shutdown sealing character-

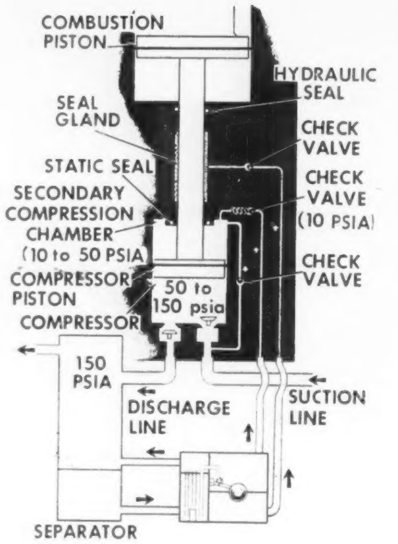


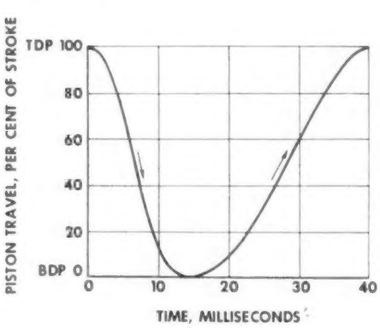
Fig. 6 Piston rod configuration of reverse leakage seal

istics, a separate static seal is used as shown in Fig. 8, with a positive O-ring seal at the top of the stripping chamber coming into action when the piston is in its shutdown position. A piston-actuated valve also cuts off the system pressure from the sealing gland oil.

A promising possibility is the use of a sealing fluid having refrigerant solubilities well below those in paraffinic-base oils. Fluids have been investigated in the laboratory having as low as 1/20 the refrigerant solubility of the oil. For any given condition of stripping temperature and pressure, the rate of refrigerant loss would be only 1/20 the value obtained with refrigerant oil, and the system would be, for all practical purposes, totally sealed.

Dynamics – Fig. 9 shows a schedule of piston motion that occurs with operation of the free-piston

Fig. 9 Piston motion of the free-piston refrigerant compressor



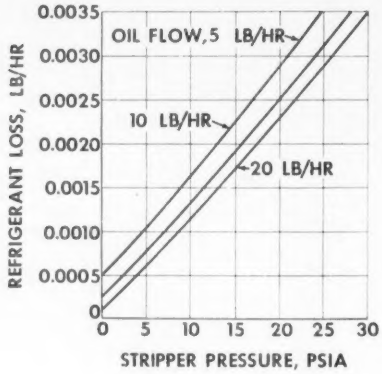
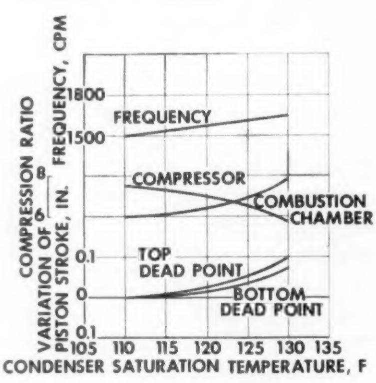


Fig. 7 Performance of piston rod seal

unit. As can be seen, the compression stroke takes only ½ as long as the inlet stroke. Therefore, for the operating frequency of 1500 cpm, the discharge valves must be designed as if they were operating in a conventional compressor operating at 2250 cpm. The design of the present compressor plenums and valving is an adaptation of a commercially available refrigerant compressor with modifications of the discharge valves to provide greater valve area.

Fig. 10 shows the limits of piston travel as condenser temperature is increased. The free-piston compressor is, in reality, a spring-mass system with the spring rate being determined by the combined pressure characteristics of the various chambers. An increase in the condenser pressure produces an increased spring rate of the system which results in a slightly higher operating speed. This increased speed is accompanied by a small shift in both the upper and lower dead points, the net effect being a

Fig. 10 Dynamic characteristics of the free-piston refrigerant compressor



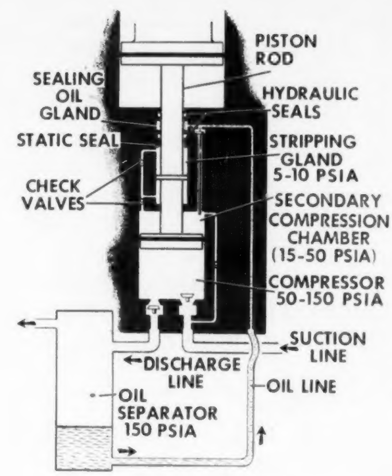


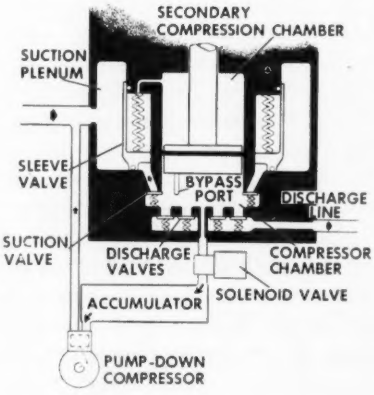
Fig. 8 Simplified piston rod seal

negligible decrease in the stroke of the unit. One characteristic of the free-piston compressor is its ability to ingest liquid refrigerant without ill effects. The liquid acts to decrease the compressor clearance volume which shortens the piston stroke.

Starting–The starting stroke of the free-piston compressor is actuated by the refrigerant compressor cylinder. When the thermostat calls for cooling, the pump-down compressor, a ⅛-hp unit, starts and evacuates the compressor cylinder. When reduced pressures are reached, the piston is drawn down into the compressor cylinder.

As the piston nears its lowest position, by-pass ports in the cylinder wall are uncovered which interconnect the compressor and secondary chambers. These ports are low enough and are not uncovered

Fig. 11 Redesign of the pump-down starting system



during normal operation. The secondary chamber is then also evacuated to its normal running minimum pressure. When the control box receives a signal that the secondary pressure is low enough to provide good starting dynamics, a quick-acting solenoid valve is opened in the compressor inlet line and the pump-down compressor is stopped. Refrigerant entering the compressor cylinder through the compressor inlet valve actuates the piston on its first upward stroke.

Fig. 11 shows a redesign which replaced the solenoid valve with a sleeve valve to close off the compressor suction ports during shutdown. Secondary pressure is exerted on one side of the sleeve valve, and when the desired low secondary pressure is reached, differential pressure automatically snaps the starting refrigerant charge to the compressor cylinder. The second design change is the insertion of an accumulator between the pump-down compressor and the compressor cylinder. The accumulator, which is kept evacuated by the pump-down compressor, is opened to the compressor cylinder when the thermostatic signal opens the small solenoid valve. Evacuation of the compressor cylinder is so rapid that small leaks in the compressor discharge valve have negligible effect on the starting capabilities.

The record of starting reliability is extremely good. On those occasions, however, when the unit does not start on the first cycle, a mis-start is sensed by the position of the starting valve which automatically closes after the mis-start, and the starting process automatically recycles.

At the present time, a mechanical starting system is being developed to eliminate the need for a pump-down compressor, a starting valve, and most of the control elements. This mechanical starting system is considered an approach to the ultimate in simplicity and reliability.

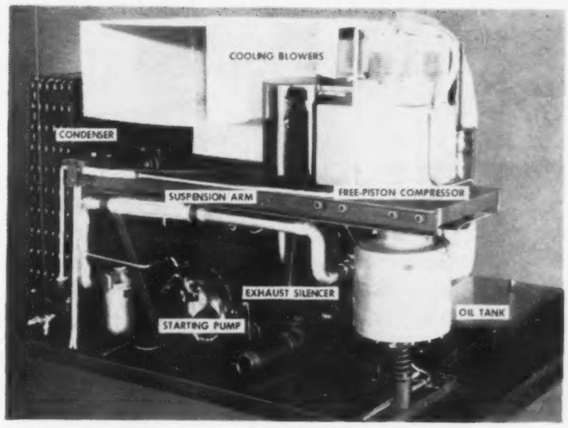


Fig. 12 Free-piston refrigerant condensing unit

Production — Fig. 12 shows a complete condensing unit which represents the present developmental status of the free-piston compressor system. Individual components of the system are indicated, including a center of percussion suspension system for the compressor, exhaust and intake silencers, condenser and engine cooling blowers, oil supply reservoirs, controls, and the presently used pump-down starting compressor. The total size of this existing unit is 50 x 33 x 30 in.

During the rapidly advancing development program, further simplifications in this system can be expected. The pump-down compressor will be replaced by a mechanical system, the cooling of both the engine and condenser will be accomplished by one blower and the compressor sealant reservoir will be built into the condenser.

A thorough analysis has been made of the anticipated cost of manufacturing a gas free-piston air conditioning system. The primary objective is to predict first cost relative to an equivalent electric air conditioning system, and, therefore, evaluate the economic feasibility of the free-piston compressor. Despite the operating cost advantage anticipated for the gas free-piston unit, marketing success requires a competitive first cost.

The electric and gas free-piston units which were compared were both of 3-ton capacity, had fan-cooled condensers, and were remote installations with the evaporators in the furnace. The free-piston unit was assumed to have a mechanical starting system, and the other improvements which reasonably can be expected to result from further development. For purposes of this analysis, costs were estimated at the system fabricator's level. Known prices of readily available components, such as condensers, fans, and electric compressors, were applied where possible.

However, detailed costs of unusual or unavailable components were carefully analyzed on a part-by-part basis. With an anticipated production rate of 15,000 free-piston compressors annually, the cost of the gas free-piston air conditioning system should be only $15 more than the equivalent electric system. An increase in production to the range justifying higher tooling investments should more than make up the cost difference.

WHO'S WHO IN ASHRAE

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See page 80, July JOURNAL

Methods and systems we use for the

Control of contaminants

In the manufacture of pharmaceuticals and chemicals there are an unusually large number of different contaminants which may be deleterious to personnel, products, building and equipment.

There is a wide variety of dusts; there are mists which are quite small droplets of material formed by violent chemical reactions; there are vapors which are a gaseous phase of substances that are liquid in their commonly known state, such as benzol, acetone, amyl acetate, butyl and methyl alcohol, dichlorethylene, etc.; there are gases which are normally formless fluids, such as carbon dioxide, sulphur dioxide, hydrogen, ammonia, etc.; there are fumes formed by distillation or chemical reaction.

THERE IS A NEED FOR CONTROL OF CONTAMINANTS

Primary reason for the control of contaminants is to safeguard the "on-the-job" health of the workers.

Although some dusts, vapors and gases may be non-toxic, it is advisable to control them to provide an acceptable environment for the worker.

Then there are the interests of good housekeeping. These are important for several reasons. First of all, the quality of the products being manufactured in a contaminated area may be affected through the lack of control. Secondly, corrosive vapors and gases, if uncontrolled, can materially shorten the life of a structure and manufacturing equipment.

We should not overlook the importance of controlling dust, vapors, and gases to minimize fire and explosion hazards. Insurance

P. J. MARSCHALL
Member ASHRAE

underwriters well know of the possible losses due to the lack of control, and, consequently, specify minimum standards of ventilation. If an owner does not meet these standards, he may not qualify as an insurable risk.

Last, but certainly not least, is the control of air pollution. Air pollution means the presence in the outdoor atmosphere of one or more contaminants such as dust, fumes, gas, mist, odor, smoke or vapor in quantities, of characteristics, and of duration such as to be injurious to human, plant or animal life, or to property, or unreasonably interferes with the comfortable enjoyment of life and property by plant neighbors.

DESIGN OF CONTROL SYSTEMS

The first question the design engineer should raise before undertaking a contaminant control system is, "Can the contaminant be eliminated, or can less toxic or less obnoxious chemicals be used in the process?" The ventilation engineer, drawing from his past experience, can estimate the approximate cost of the contaminant control system. This estimated cost should be given to the development laboratory section and to the research scientists so that they may probe the costs of research and development to find substitute chemicals or a new process.

If it does not appear practical to spend the time of the research and development section to develop a new process, the next step is to see what can be done in the way of containing the contaminants in order to minimize the ventilation requirements.

If the contaminants originate at scattered points throughout the manufacturing area and it is determined that it is not practical to collect these contaminants at the source, the dilution method of control is sometimes applicable. However, generally speaking, the preferred method of controlling contaminants is to collect them at the source. The dilution method may be an auxiliary requirement of the insurance company to dilute flammable vapors in case of an accidental spill.

The quantities of air required for local exhaust systems are based on several factors, the most important of which is capture velocities. For most applications in our industry, these velocities will range from 50 to 200 fpm. The lower capture velocity is used to control contaminants released at low speed in relatively quiet air. The higher cap-

Contaminated air may be injurious to workers, deleterious to the manufactured product and costly to plant operations. It is essential that contaminant control systems be effective and economic. In this study, the author discusses various systems chiefly related to pharmaceuticals but, more broadly, the basic problems involved elsewhere, too, and their possible solutions.

P. J. Marschall is Manager of the Engineering Div, Abbott Laboratories. This paper is a somewhat condensed version of "Control of Contaminants" as presented at the Industrial Ventilation Conference at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959. The full text will appear in a Symposium Reprint covering all papers presented at this Conference.

ture velocity is used to control contaminants released at a high rate or released in an atmosphere of disturbing cross currents. Other factors are area of openings and capture distance.

TREATMENT OF EFFLUENT

Removing the contaminant from the occupied area is only part of the problem. We cannot as a rule discharge the effluent outdoors without treatment. There are a number of factors to consider in selecting the type of equipment for treatment of the contaminated effluent from a manufacturing operation. If the material is a dust and is valuable enough to warrant collection for recovery, we use a dry cyclone collector or a bag or cloth type collector. We also use a bag type collector for highly toxic dusts where a fairly high degree of efficiency of collection is desired. When bag type collectors are used and it is desired to maintain close control of the volume of air exhausted from a particular area, such as an air conditioned room, we use static pressure regulators to maintain the air volume reasonably constant.

If recovery of the contaminant is not economically practical or desirable, and the contaminant is not highly toxic, a wet type collector or scrubber is generally used to collect or absorb the contaminant. When wet type collection is advisable we use the wet type dynamic precipitator or the wet centrifugal type. However, if the contaminant is present in large quantities, or if it is toxic, it is quite likely you will not be permitted to discharge the effluent from the wet type collector into the storm or sanitary sewer or into any stream, river, lake, creek, or other body of water without treatment.

In our plant, in addition to the storm and sanitary sewer system, we have a chemical sewer system which discharges into our own chemical waste treatment plant. If the contaminant is toxic to the waste treatment process, then the contaminant must be collected, hauled to a dump area, and buried.

In treating harmful or obnoxious gases, vapors, and mists we employ several methods.

Where the chemical reaction results in the generation of hydrochloric acid, sulphuric acid, acetone vapors, sulphur dioxide, nitric oxide, or phosgene gas, we use a packed column. These columns range from 10 to 20 in. diam and are filled to a depth of 6 to 10 ft with beryl saddels or roschig rings. Water is delivered to distributing troughs or headers in the top of the column at rates varying from 10 to 40 gpm. The gases or vapors are scrubbed as they pass through the wetted column, and the effluent is then discharged to the chemical sewer.

To remove ammonia gas we use a jet type scrubber. This type of scrubber is also used for removing hydrogen sulfide. However, instead of using plain water in the jet, we use a 10% caustic solution. Here again, the effluent is discharged to the chemical sewer.

In some instances it is neces-

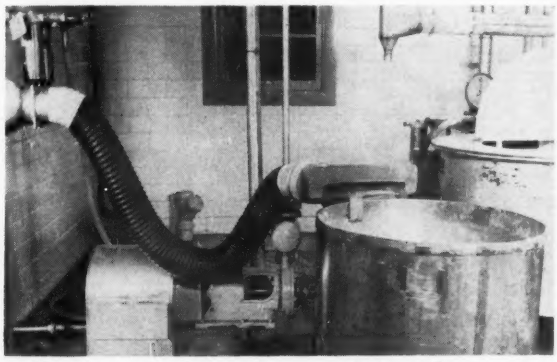


Fig. 1 Design standard used for exhausting contaminants from an open drum or tank

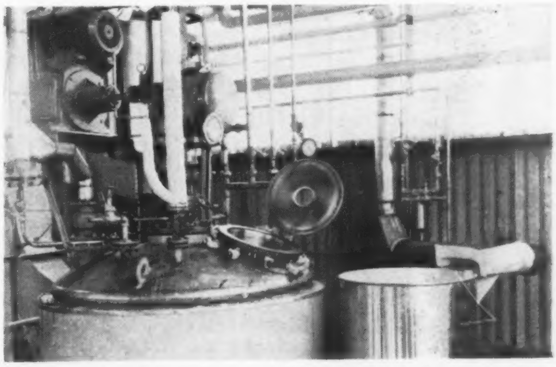


Fig. 2 The valve in this exhaust duct connection is necessary if vessel is pressurized at any given time during the process cycle

Fig. 3 Counter-balanced damper at the end of each branch assures that no air will be exhausted to the branch if it is not coupled to a tank

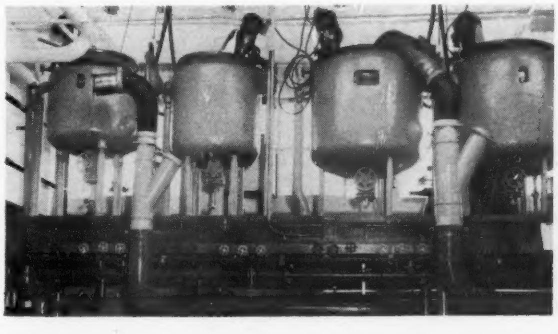
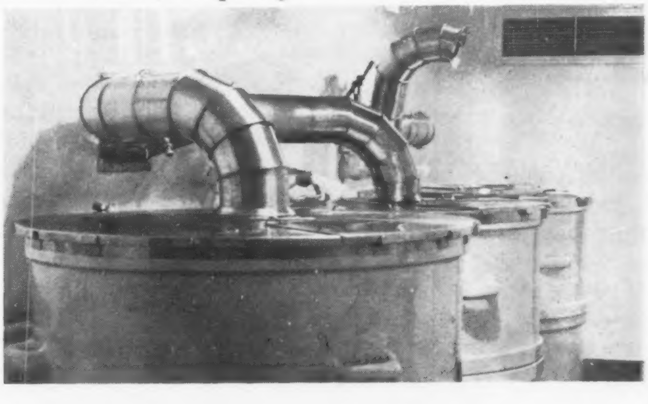


Fig. 4 Sectionalized covers permit accessibility to all of the area within the crock with minimum opening



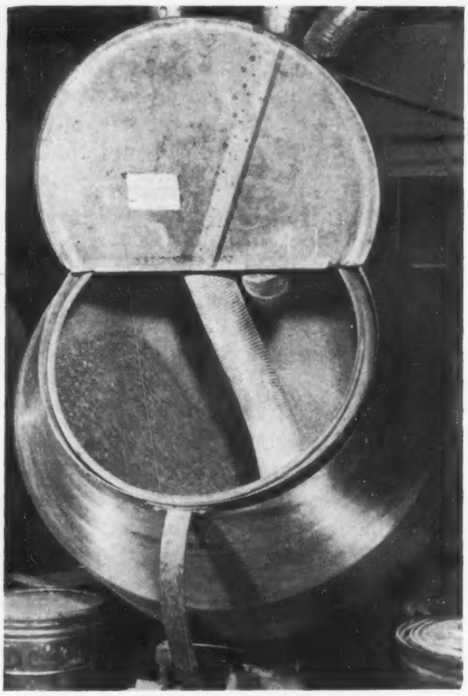


Fig. 5 Supply and exhaust nozzles in the coating pan. Pans revolve at a speed of 40 rpm

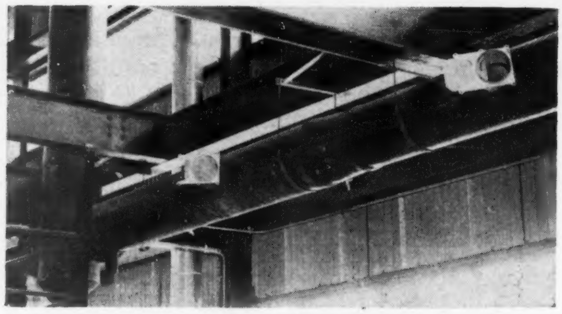
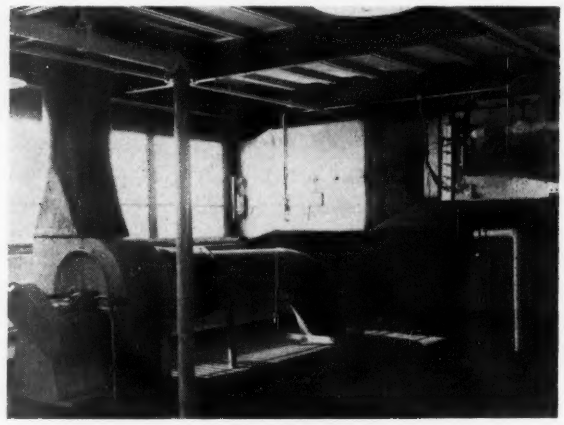


Fig. 6 Horizontal main constructed of furan plastics. Branches provided are for connections to future equipment

Fig. 7 Fan and plastics duct connect to main and discharge through the roof. At right is a make-up air unit



sary to collect the effluent in holding tanks and then discharge it to the chemical sewer at a predetermined time and rate in order not to upset the operation of the chemical waste treatment plant.

When economically justified, gases and vapors are recovered by condensing them in a closed heat exchanger.

Illustrated are some of the units of equipment and systems we have used for the control of contaminants.

Fig. 1 shows a design standard used for exhausting contaminants from an open drum or tank. This standard design form becomes a part of the permanent record of the contaminant control system. The exhaust duct in Fig. 2 is connected directly to a flange nozzle in the head of a reactor. Note the drain boot and the valve in the duct connection. The valve is necessary if vessel is pressurized at any time during the process cycle.

There are instances where a chemical manufacturing operation requires exhaust from a large number of pieces of equipment, all of which do not require exhaust at the same time. Fig. 3 shows a system exhausting a limited number of pieces of equipment at any given time. Rather than depend on operators to manipulate dampers, we supply a limited number of detachable exhaust nozzles. The nozzles, which are the quick-coupling type, can be switched from one tank to another in a matter of seconds.

Fig. 4 is an air conditioned sterile room which is under pressure at all times. The problem here is to remove butyl alcohol vapors from the filtration crocks. The sectionalized covers permit accessibility to all of the area within the crock with minimum opening. With this type of design, the exhaust required to control the vapors is kept to a minimum. The make-up air load on the air conditioning system is, therefore, also kept to a minimum.

In Fig. 5 the supply and exhaust nozzles are in the coating pan. These pans revolve at a speed of approximately 40 rpm. The first stage of sugar coating operation is extremely dusty, and is referred to as sub-coating operation. About seven or eight coats of a resin material are applied and after each coat, talc is added to prevent the tablets from adhering to each other. The problem is to keep the dust from spreading out into the room. Film tab process does not involve powders but does involve acetone vapors.

The horizontal exhaust main of Fig. 6 is constructed of a furan plastics material. This material is highly resistant to most corrosives, gases and vapors. In comparing this type material to stainless steel and galvanized steel, it is difficult to make an accurate comparison because of the wide range of bids received on plastics ductwork. The plastics material is used only for the main runs and will run about 2½ to 3 times the cost of galvanized steel. The furan resin plastics ductwork, we believe, will last for the life of the installation of the manufacturing equipment.

The exhaust fan and the plastics duct as connected in Fig. 7 discharge through the roof. The interior of the fan housing and the fan wheel are given five coats of a thermosetting epoxy phenolic material. The make-up air unit on the right supplies air to meet the exhaust requirements.

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The backbone of the publications program is the ASHRAE JOURNAL. This monthly magazine is the official publication of the Society and through its pages, the business affairs and technical activities of the Society are brought to the members and made available to the public. The JOURNAL is primarily a high-level technical publication which presents to *all* the members the best of the technical papers presented before the national, regional and chapter meetings of the Society. In addition, it carries publicity of the national meetings and reports on the important business activities. It carries news of chapter and regional organizations and meetings and for general information lists new products and Industry news. The JOURNAL also provides a unique advertising medium through which business organizations, serving this Industry, can carry their advertising messages to the leading engineers in this field.

For a more permanent record of the Society's activities, the ASHRAE TRANSACTIONS annually publishes the technical papers and discussions from the national meetings in a bound volume. Also included in this volume is a record of the actions of the Board of Directors, the membership of all boards and committees and the annual reports of major committees. ASHRAE TRANSACTIONS thus provides a permanent record of all of the technical and business activities of the Society considered to be of lasting value.

Prior to the merger, each of the former societies published an annual reference book of engineering information and technical data. The ASHAE GUIDE and the ASRE DATA BOOK were both well known and highly considered. As a result of the merger, ASHRAE will now publish the ASHRAE GUIDE AND DATA BOOK, combining the best features of each and expanding the coverage of both. The new reference book will be published in three separate volumes. Vol. I, to be published in 1961, will cover all types of Refrigerating, Heating and Air Conditioning Equipment. Vol. II, to be published in 1962, will cover all types of Refrigerating, Heating and Air Conditioning Applications. Vol. III, to be published in 1962, will cover the Fundamentals and Tables for these fields of engineering. These three volumes will give every member the finest reference library of engineering information available. Vols. I and II will be kept up-to-date with new issues published on alternate years.

A more detailed coverage of current engineering developments in special fields of interest is provided by the Society's "Symposium and Conference Bulletins." These publish full and complete reports of the many engineering Conferences and Symposia held at the national meetings. Because of the high caliber of the participants and the broad coverage of all facets of a selected subject, these bulletins can be of great value to the engineers active in the fields covered.

As a result of the active research program directed by the Research and Technical Committee, the Society, from time to time, publishes Research Reports. Frequently, the results of a research project are presented in one or more technical paper and published in the JOURNAL and/or TRANSACTIONS. Complete reports of important projects are published as separate Research Reports and are available from Society headquarters for a nominal charge.

As new engineering standards are developed by the Standards Committee and approved by the Board of Directors, they are published by the Society and made available to the members and the Industry for a nominal charge.

Another Society publication of interest to each member is the Membership Roster. Here, every member is listed with his mailing address. This provides a means for members to keep in touch with others who are working in their field of interest.

The Publications Committee is responsible for all of these publications. It is the objective of this committee to be as responsive to the needs and desires of the members as possible. Each of these publications is intended to provide a specific service to the members of the Society. They are published for you and except as indicated above are sent to you without charge as a part of your membership service. How do you like them? Do you have suggestions for improvement? The Publications Committee solicits your comments.

Some refrigerators

are still too noisy

E. A. BAILLIF
Member ASHRAE

J. P. LAUGHLIN

Modern household appliances probably are receiving more attention now from the standpoint of noise than they have ever before. It is a never ending battle to keep this noise within acceptable limits and of all appliances, the battle rages more furiously over the refrigerator than any other; quite probably because its main job is to be cold, something which has no noise-producing connotations to anybody. Due to a moderately successful waging of this never ending battle, most of them do their job unobtrusively. This paper describes the processes involved in the pursuit of the solution to the many, many problems associated with refrigerator noise control.

Tools for testing – The first essential, even if the "instruments" are ears, is a sound test area. This, for some evaluations, might be a quiet corner of the laboratory with makeshift walls or heavy curtains to form a "listening booth." But this becomes quite inadequate for investigation of noise contributions in most frequency bands, and for establishing a history of reproducible data, a specially isolated room is a basic requirement. This room may be either anechoic (free field) or reverberant (diffuse), but one which has some of the characteristics of both is probably the most common. This compromise room is anechoic at notably high frequencies and essentially reverberant at low frequencies.

In almost all cases where the laboratory is in an industrial environment, a double wall, double door room is necessary. The inner room is suspended on either springs or rubber to isolate all frequencies above about 4 cps. This is a fairly practical limit and if a room is designed to isolate much below this, it becomes "wobbly" and uncomfortable in which to work.

The shape of the inner room is a factor, due to standing waves. Generally speaking, a rectangular room is used and to provide a good standing wave ratio, i.e., the most diffuse field, its dimensions should be in a ratio of 3-4-5.[1]

The facilities here described are those of the Evansville Div of Whirlpool Corporation. In these facilities there are at present several sound test rooms available for product engineering test of refrigerators or their components, all of the "padded cell" or semi-reverberant type.

A double wall room consisting of one padded cell and one reverberant test cell is under construction presently. One part of this, the padded cell, is complete and in operation and is shown in Fig. 1. A schematic of the complete arrangement is shown in Fig. 2. It can be seen that the instrumentation is outside the suspended test chamber, a good arrangement, because otherwise the instrument noise may interfere with the test in progress. However, this has the disadvantage of removing the operator from the "feel" of the object on test.

Special attention must be paid to the openings in the rooms. Each wall should have its own door and both of these doors must have an effective door gasket seal. All other openings for power, lighting, instrument cables, etc., should be well caulked. Any narrow aperture will create an acoustic leak, especially at high frequencies.

Temperature control of the test cell and instrument areas is desirable both from the standpoint of operator comfort and for minimizing the temperature effect on the instruments and the operating noise of the test object. The least expensive method is to provide a direct expansion, fan cooled refrigerated coil in the test room with an external refrigeration system,

"The best way to get rid of noise is to design it out," conclude the authors. Further, "basic instruments are really only the starting point in analyzing noise and can be supplemented by special apparatus, limited only by the imagination of the engineer, to attack any phase of the compressor as a noise generator." An example of a special test method used in this study is found in the use of magnetic pick-ups on a rotating part of the compressor in combination with a capacitive pick-up formed from the discharge valve of the compressor, to investigate the effect of compressor valve action on the noise spectrum.

[1] According to Dr. H. F. Olson, RCA Laboratories.

Dr. E. A. Baillif and James P. Laughlin are members of the Product Engineering Dept, Whirlpool Corporation. This is a condensed version of "Investigation and Control of Refrigerator Noise" as presented at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959. The complete paper will appear in ASHRAE Transactions.

which is then pumped down and the fan turned off for complete absence of noise during test. This type system functions during non-test periods with the room doors open to provide ventilation. Fig. 3 shows the arrangement for air conditioning the room in Figs. 1 and 2 which utilizes an air conditioned outer room with duct treatment to minimize noise carry through, and with a baffled low velocity conditioned air system continually changing the air of the test cell.

In this test room the practice has been to endeavor to make the refrigerator perform noise-wise in a manner similar to that encountered in a home. To this end, a solid floor is usually used under the refrigerator on test. Generally, one wall at least is also hard surfaced to give the effect of the reflective plaster or tile kitchen wall, while the remainder of the interior surface of the test cell is treated with acoustic material to provide an environment with controlled absorption.

The science of sound is not a simple subject and its study must involve not only physical factors, but psychological factors as well. This is because the human ear is one of nature's more complex structures, and not only functions as an extremely sensitive microphone, but also as a selective analyzer with the ability to extract pitch, loudness and "quality" of noise in the presence of interfering background noise.

The task then of finding an instrument to make physical measurements of noise in some manner so that some of the human reactions to it can be classified, would seem a formidable one. Besides this, the factors mentioned above are interdependent, and also are affected by the experience of the listener.

There are basically two objective properties of noise which are measurable. These are sound pressure level and frequency, and relate subjectively to loudness and pitch. Therefore, to attempt to instrumentate and achieve the measurable objective properties of noise there must be provided a transducer to pick up air borne or structure borne vibrations since noise has its source in vibration, that is,

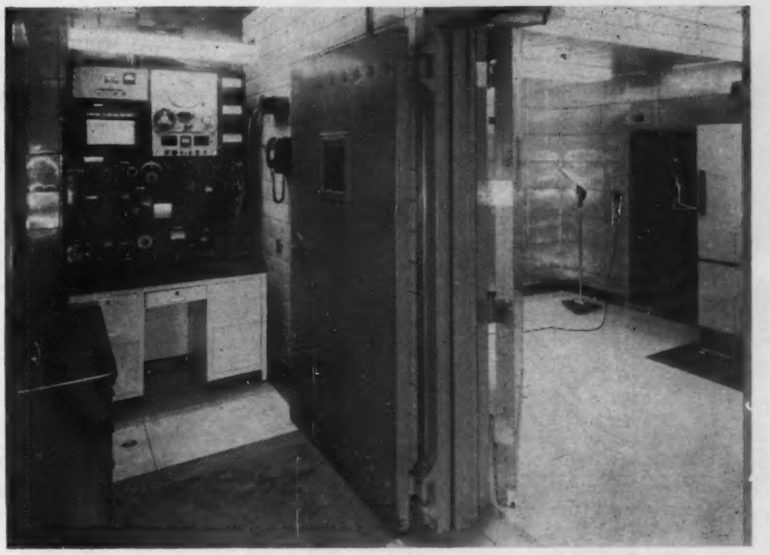

Fig. 1 Engineering sound room

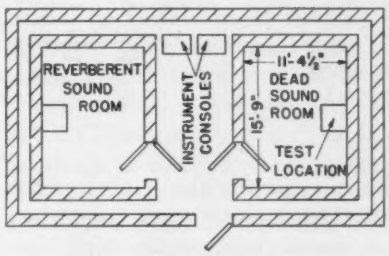

Fig. 2 Schematic diagram of sound rooms

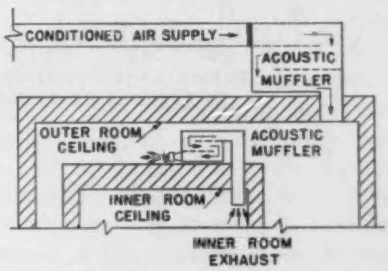

Fig. 3 Schematic diagram of air conditioning system in sound rooms

a microphone or vibration pickup; an amplifier to transform the output of the transducer into a useful signal level; a means of controlling the signal both as to intensity and choice of frequency response, namely, an attenuator and weighting networks; a means to segregate out certain specific frequency ranges for detailed analysis; and a means of indicating or recording the output signal.

The commercial forms of these instruments are described in many places in the literature and will not be discussed further here.

Interpretation of results — Almost universally, when "noise measurement" is mentioned, the immediate thought is "decibels." Actually, the use of a single value as a rating for the loudness of a noise has a serious drawback, namely that decibel readings have little relation to the "loudness" of a noise as "heard" by the ear. This is because the ear's response is not flat with respect to frequency, or in other words it tends to evaluate different frequencies differently. Although this is partly taken into account by weighting networks on the sound level meters, it is not nearly accurate enough to make the resultant db readings represent a rank order of loudness.

Much more satisfactory is a rating system based on loudness in units of sones which does not pre-suppose similar frequency distribution. To obtain the sone value for a sound, it is first necessary to feed the flat network signal from a sound level meter into, for example, an octave band analyzer, which gives a db reading for each of a series of frequency bands. From a loudness vs. octave band sound level chart, can be obtained the sone value for each frequency band, and by then applying a simple formula, the overall loudness in sones for the specific noise in question can be obtained readily.

There are two methods of performing the above mentioned computation of sones, each based on its

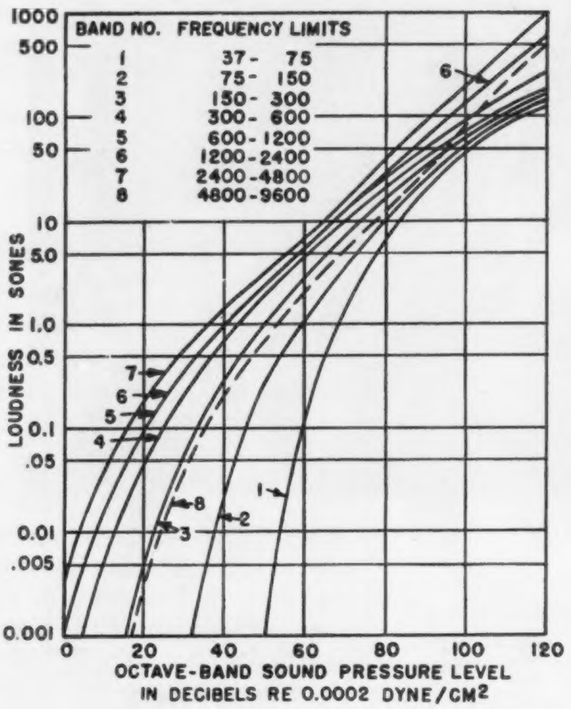

Fig. 4a **Chart for converting from octave band levels in db to loudness in sones (after Beranik and Peterson)**

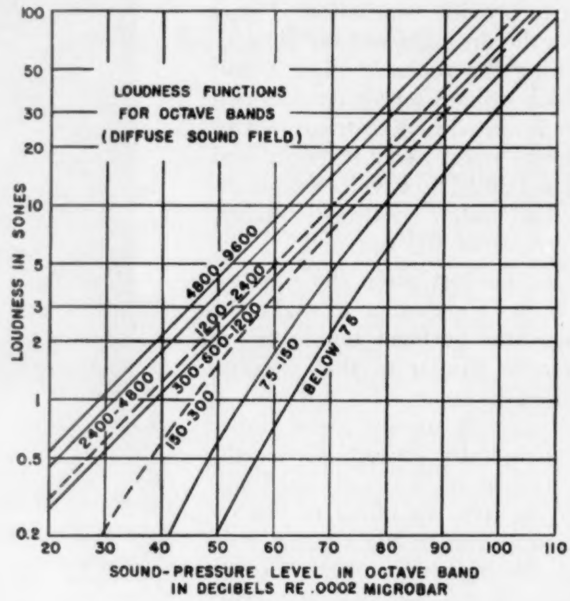

Fig. 4b **Chart for converting from octave band levels in db to loudness in sones (after Stevens)**

own chart and formula. The one which has been in use for a number of years and is still a useful method is that which utilizes the chart in Fig. 4a which is based on work done by Beranek and Peterson.[2] After the sone value is determined for each band the total number of sones is obtained by simply adding together the loudness for all bands, or:

$S_t = S_1 + S_2 + \cdots + S_n = \Sigma s$
Where: S_t = overall loudness
$S_1, S_2, \cdots$ = loudness in each band
Σs = sum of all loudness for all bands

The other method has been rapidly gaining favor in the literature and in practice and is from the work of Dr. S. S. Stevens of the Psycho-Acoustic Laboratory at Harvard University. This consists of using the chart shown in Fig. 4b,[3] and than applying the formula:

$S_t = Sm + F(\Sigma s - Sm)$
Where: S_t, Σs are as before and
Sm = the loudness of the loudest band
F = a factor depending on measurement conditions (0.3 for octave band measurements).

It can be seen that it is possible by either method to have a sound with a lower decibel reading than another, be the louder of the two. This is quite often true when one sound has strong high frequency components as compared to another which has strong low frequency components. The one with the predominent high frequency will generally be the louder even though the db ratings may be equal or even reversed.

Method of testing compressors — Since the modern refrigerant compressor is basically a high speed electromechanical device, noise reduction measures would be expected to and do require the use of many techniques involving vibration isolation, pulsation mufflers, flexible connecting tubing, and component balancing. However, due to the thermodynamics involving relatively large pressure and temperature variations, oil refrigerant mixtures, and the miniaturization of components, there are other special techniques demanded over and above those encountered in noise studies involving a less complex mechanism.

For example, where the noise reduction is concerned with the metallic parts of the mechanism, one must consider surface finish, clearances, and assembly relationships. Although these are held to extreme tolerances and materials must always meet carefully established specifications, they can still be a factor in the resultant noise when altering the design to accommodate a different unit or to effect cost reduction. Admittedly, the noise here is controllable by emphasis on quality of parts and assembly procedures, once the design is established, but the procedures involved in determining the economical limits on quality, as it affects noise, call for special investigative processes.

One method which is reliable but costly is the exhaustive method of testing each variation in design in steps to establish a trend or a breaking point. A method leading to the same result, but with less time and effort is preferable. This is possible when one departs from the basic sound level meter and endeavors to analyze the frequency components of the compressor noise. To accomplish this, one can make use of auxiliary test apparatus including frequency analyzers, pressure transducers, magnetic excitation of components and possibly most importantly, the science of mechanical vibrations itself.

Of special interest, from a design standpoint, are resonant frequencies of machine components. These can sometimes be calculated but because of the complexity of

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[2] Willem Brand and Jens T. Broch, Bruel and Kjaer Technical Review (Brush Electronics Co.) No. 4 1956 Fig. 4.

[3] S. S. Stevens, Noise Control 3, 11 (September 1957)

Fig. 5 Compressor arranged for noise test on a spring isolator

Fig. 6 Quality control noise test cell for compressors

the shape of the component and the manner in which it is attached to the system this usually is difficult if not impossible. However, on a sample part, the resonant frequencies can be determined by either striking it or driving it electromagnetically while comparing the emitted tones with a variable audio frequency sound source. Resonant components are bothersome because they have families of resonant frequencies, namely, multiple harmonics, any one or several of which may be excited into forced vibration either by repetitive impacts or oscillatory exciting forces near resonance.

With all the defects that are possible, which can also influence performance, it is easy to see why high standards are set on the production of parts for compressors. Long experience and constant surveillance by quality control allow few compressors to reach final compressor assembly with adverse noise, and it is at this point that a noise check is made. This is the so-called "free space test" wherein a compressor is run under normal operating conditions within a noise test cell. The "free space" noise test takes two forms as shown in Fig. 5 and Fig. 6. Fig. 5 shows the compressor arranged for noise test on a spring isolator in one of the product engineering sound rooms. The engineering test makes possible the analysis of new compressor designs and also the spot checking of production compressor noise. Fig. 6 shows the quality control noise test for compressors and is used as a gauge of production noise. Compressors whose noise levels exceed a standard are examined and the deviations found referred back to production to permit rapid correction of the production process responsible for the noise producing deviations.

Having traced the compressor through production, let us retrogress to the design standpoint for a further look at the methods of analysis of noise, and some examples of the results obtained.

Engineering noise tests based on theory, assumptions, and the actual insertion of known defects into the compressor have been made to build up a backlog of valuable comparative data. Thus, when a test is run with results as shown in Figs. 7 and 8 (which show the "A" network and octave band readings in decibels, and the computed sone rating, by both the Beranek and Stevens methods, for an experimental compressor tested under operation on the engineering free space set-up), it is possible to compare these data with the reference data from prior tests. By proper interpretations of these data it is to some extent possible to determine whether this compressor will perform satisfactorily from the standpoint of noise when applied to a refrigerator. It should be noted that proper interpretation is important since differences in the test set-up and sound reduction measures used on the refrigerator cabinet alter certain frequency bands when the compressor is applied to the cabinet.

To further pinpoint the noise spectrum a frequency analysis is also made, an example of which is shown in Fig. 9. This is a discrete frequency spectrum for the same compressor shown on an octave band basis in Fig. 7. It can easily be seen how the discrete or individual frequencies can be identified by this method and a more detailed analysis thus made possible. Other frequency analyzers which present a continuous portrayal of amplitude vs. frequency are also available.

A still further step can be made in the analysis by recording on tape the effects of either component or operating condition changes and then playing back short connected excerpts of these tapes for aural judgments. This is especially helpful when delays are encountered in experimental changes, or it is desired to compare experimental designs vs. production on an aural "before and after" basis.

Thus far only the airborne vibration transducer or microphone

Fig. 7 Compressor noise expressed in db

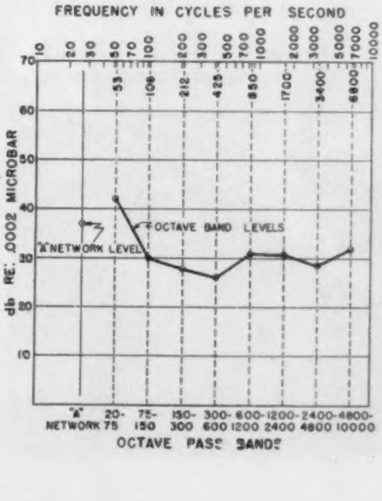
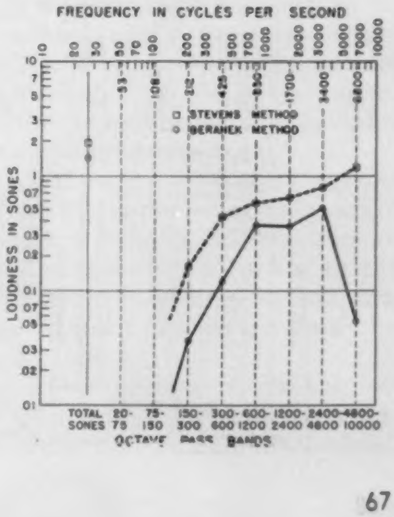


Fig. 8 Compressor noise expressed in sones



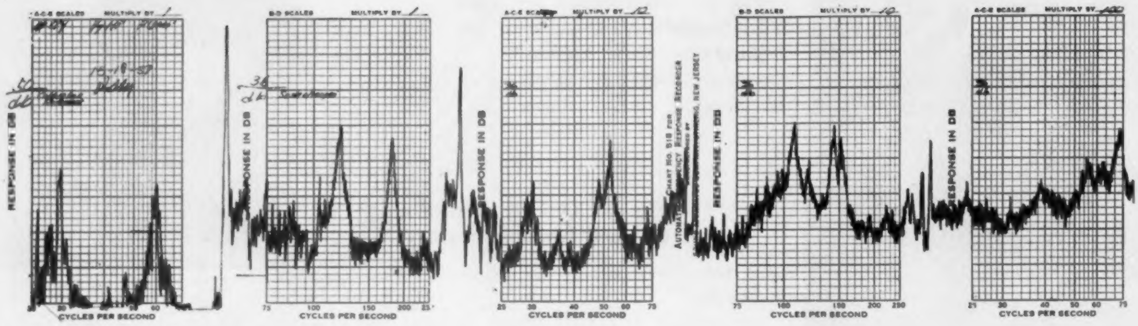


Fig. 9 Frequency chart of refrigerator compressor noise

method of analysis has been discussed. There are other transducers which also will contribute valuable information on noise analysis, for example, the vibration pickup. Use of these in the torsional, vertical and horizontal aspects of vibration of the compressor can be helpful in the determination of component unbalance and pressure unbalances in the compressor. Compressor housing vibration patterns can be traced out with a low mass pick-up placed successively at multiple grid positions on the housing.

A device especially valuable in diagnosing compressor noise and the resultant unit noise is the pressure transducer. Applied to the gas stream of the compressor this transducer can show the effects of internal component changes on the pressure wave and the before and after effects of mufflers. Fig. 10 shows a comparison of the data available by means of the above test methods for noise, torsional vibrations, and pressure pulsation on an experimental compressor.

The possible tests have by no means been exhausted at this point as two further examples will show. One interesting method of analysis is that of isolating one disturbance by removing its effect from the field of the compressor noise. An

Fig. 10 Frequency charts of compressor noise, vibration and discharge pressure

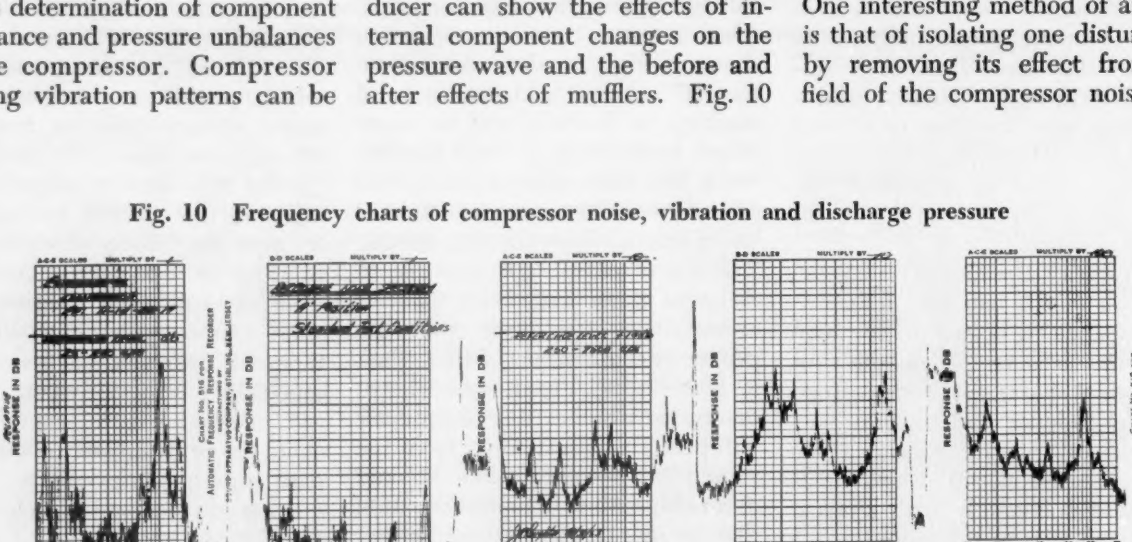


Fig. 11 External drive compressor test stand

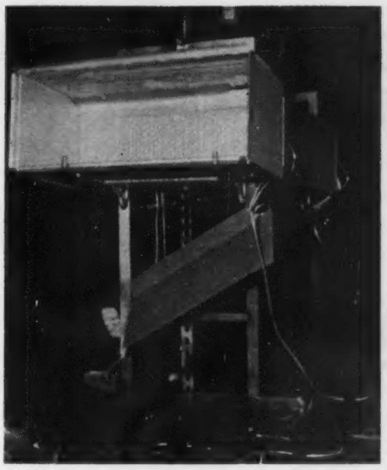
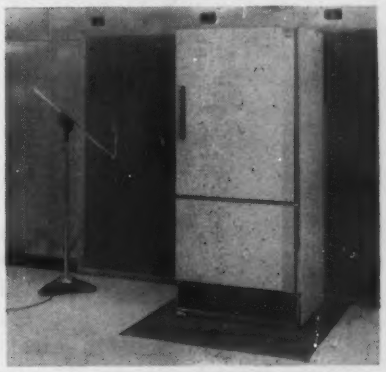


Fig. 12 Quality control acoustical labyrinth for unit noise test

Fig. 13 Refrigerator in engineering sound room



example of this is shown in Fig. 11 which shows an external drive compressor test stand. This permits the compressor to be driven either by its motor, or by a motor coupled through a long shaft to the compressor.

A further example of a special test method is found in the use of magnetic pickups on a rotating part of the compressor in combination with a capacitive pickup formed from the discharge valve of the compressor, to investigate the effect of compressor valve action on the noise spectrum.

It should be apparent that the basic instruments are really only the starting point in analyzing noise and can be supplemented by special apparatus, limited only by the imagination of the engineer, to attack any phase of the compressor as a noise generator.

Method of testing units—The type of instrumentation chosen for a specific noise evaluation depends ultimately upon the purpose for which the information is to be obtained. The human ear still has its place in the evaluation of noise if the observer is trained for this purpose. A working example of this type of evaluation is shown in Fig. 12 which is an acoustical labyrinth used on the production line to 100% check unit noise.

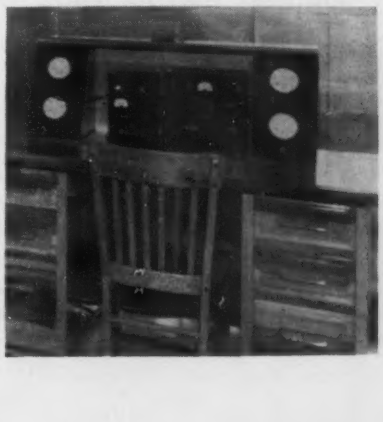
It is not feasible economically nor necessary to 100% noise test refrigerators at production rates common in refrigerator manufacture. However, potentially noisy units should be culled from the unit assembly line to avoid the cost of possible later rejection as a finished refrigerator, or a possible field service call.

The labyrinth serves this purpose by providing a low ambient noise level to enable a trained operator to aurally evaluate the merit of each compressor as it passes him on the moving conveyor chain.

One can appreciate well that in any system of this sort there will be some conveyor noise, rattles of evaporator tubing, etc., but the trained ear can easily discriminate against this extraneous noise as long as the ambient noise is held to a low level.

It should also be mentioned that the ear is not infallible due to the human element, and in the case of a high percentage of rejected units they are rerun and listened to by a committee to determine whether the compressor assembly has a potential problem needing immediate attention, or whether the operator just had an off day.

Fig. 14 Instrument console in quality control sound room



The actual percentage of rejects due to this test is quite low by reason of the high standards set on compressor assembly and the aforementioned compressor noise test, but it serves as a double check on unit noise and promotes a high quality standard.

Testing complete refrigerator—This is probably the most important test since now for the first time all the potential noise producing components are in place.

The first consideration of the overall refrigerator noise, of course, occurs in the design stage, then the engineering sound laboratory proves the design and it is placed in production. Here again the overall noise must be evaluated to insure that appurtenances, materials, assembly procedures, etc., are not contributing to offset the careful noise screening already accorded the compressor and unit.

Those methods used to accomplish this in the industry are varied; ranging from visual observation of the components to 100% testing of all refrigerators by a method similar to that described previously for unit testing.

One method which has met the criteria of high quality at reasonable expense is to 100% visual-

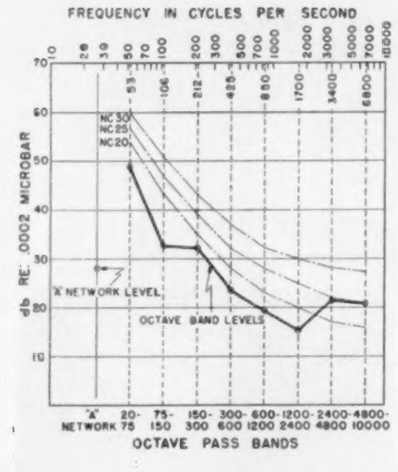


Fig. 15 Refrigerator noise expressed in db as compared to noise criteria curves

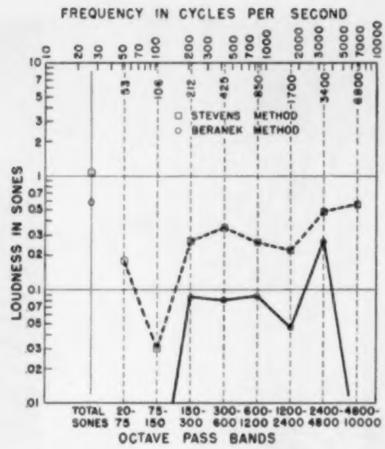


Fig. 16 Refrigerator noise expressed in sones

ly check the refrigerator for assembly defects, test a quality control sample lot before crating, and a lesser number of samples from the warehouse. The latter test is sometimes conducted by the engineering sound laboratory and permits including the processes of crating, transport, and uncrating, as of course would be the case in the store or customer's house.

Again, as previously mentioned, much can be said as to how to noise test the refrigerator; one method which has been time proven is hereinafter described. The actual testing arrangement is shown in Fig. 13, which is a view of a refrigerator situated in place in the engineering test room. It should be mentioned at this point that the quality control sound test room and the test method used are identical to that used by engineering except for the elaborateness of the instrumentation. The quality control sound room instrumentation is shown in Fig. 14.

The refrigerator is run, usually outside the test rooms, for an interval to reach equilibrium conditions, before any sound readings are taken. This is done in order to achieve a more representative operation than would occur during a pulldown.

The microphone and the measuring instruments comprise the balance of the test set up. The microphone is placed at a height of 48 in. from the floor, which is the approximate ear height of a seated person and at the three positions around the refrigerator, the three positions providing a means of arriving at a more average level.

Data from these tests are obtained from the various instruments as described here. The sound level meter provides "A," "B," and "C" network readings. Background noise level corrections, if necessary, are made on the readings and the three "A" network readings arithmetically averaged and this figure recorded. For years this was the extent of a noise test in the industry and as mentioned before, as long as one refrigerator is compared to another both having similar frequency spectrums, this average decibel reading corresponds reasonably well to a loudness rating in sones. Sone readings, obtained as previously described, are now used, as well as the noise criteria curves,[4] but it has so far been impractical to completely eliminate the "A" network reading due to the long association with it by people who have only limited contact with the noise test process.

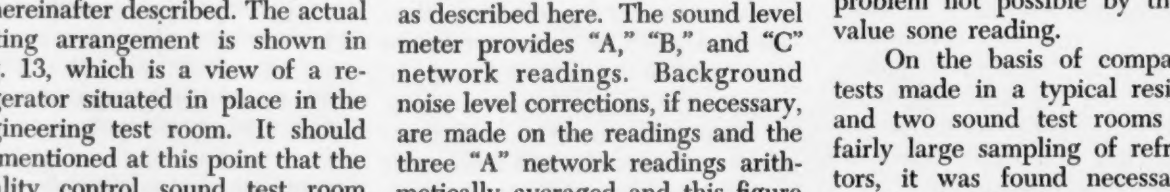
The noise criteria curves are rapidly gaining acceptance in the literature and have served as a basis for establishing a go-no go criteria for the quality control sound room. Their particular advantage, from this standpoint, is the octave frequency display, permitting the localizing of a noise problem not possible by the one value sone reading.

On the basis of comparative tests made in a typical residence and two sound test rooms on a fairly large sampling of refrigerators, it was found necessary to modify slightly the noise criteria curves as published, in order to accommodate the acoustical environment of the sound test rooms.

To obtain the data necessary for the sone and noise criteria report, the output of the sound level meter is fed into an octave band or similar analyzer, for the three microphone positions; the decibel

[4] L. L. Beranek, Noise Control, Vol. 3, No. 1, 19-27, January, 1957.

Fig. 17 Frequency chart for refrigerator noise



level in each band obtained is corrected for background noise, if necessary; and each octave band set of readings is arithmetically averaged. From this data the noise criteria plot can be made directly, and as previously described, a sone value computed. A typical set of data for a refrigerator is shown in Figs. 15 and 16.

This is the extent of the quality control noise test as far as obtaining and reporting of the individual datum is concerned. Engineering has, however, a somewhat broader function, and hence extends the test data gathering to include the use of some additional equipment.

Thus, the output of the sound level meter can be fed into a discrete frequency sound analyzer and recorder; this is usually done for one or more positions of the microphone. Such a record enables one to examine individual frequencies and hence predict or theorize with much greater accuracy what is causing a specific noise. A typical chart from one such analyzer-recorder system is shown in Fig. 17. This is for the same noise as was shown in the form of octave bands in Fig. 15.

Of a somewhat different nature but also of considerable value is the sound recording on magnetic tape of the refrigerator noise. By this method a design can be checked before and after a change is made; designs can be compared to competitive refrigerators; a noise test can be made in the field for later analysis, etc. The value of this test method, which supplements those described previously, is the obtaining of a permanent historical record which can be analyzed again at a future date should it be desired to re-evaluate any data. It should, of course, be noted that the tape system should be of the highest quality since the inaccuracy of the recorder, however small, is superimposed on the inaccuracies of the other instruments.

Again, there are several other devices which may have a special application such as the oscillograph, impact noise analyzer, strobe lamp and vibration meter, etc. Although use is made of these instruments in the engineering laboratory as occasions may dictate, they are elsewhere discussed in the literature

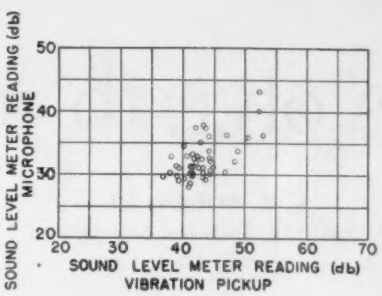


Fig. 18 Refrigerator sound level vs. cabinet vibration

and with one exception, to elaborate on their use here is beyond the intended scope of this paper.

The one case discussed here is an interesting possibility which was closely examined several years ago. The thought was that a vibration pickup, which could be used on the assembly line might be applied at a predetermined spot on the compressor, unit, or cabinet, so located as to produce an indication which would correlate to the sound level of the machine. If this could be done an especially valuable means of controlling noise could be obtained. Several types of pickups were investigated and many different locations explored, but to little avail. There was some correlation in the most noisy machines which would have been singled out anyway, but the correlation did not exist to a useful degree for instance on a series of cabinet tests, as can be seen from Fig. 18. The failure of such a test method quite probably stems from the fact that the stresses and vibration patterns in the devices are not closely enough repeated from one to the next to provide a valid comparison. However, the fact remains that quite a valuable noise control tool would be realized if some such device could be made to do the job.

Factors in Noise Control – It is understood generally that noise is generated and transmitted by a number of basic physical processes. And further, that it can be eliminated frequently or reduced by redesigning the noise producing devices; by placing the machine on an appropriate mounting; by adding mufflers, filters, or dampeners to absorb, block, or dampen the noise; or by modifying the device to change its noise characteristics so as to make it less objectionable.

Within this wide scope, one can imagine some more specific applications of noise control as applied to refrigerators, which generally illustrate two avenues of approach: one where the same device or change can be made to accomplish some additional desirable function, the other where a large significant noise reduction is possible. The guiding principle here is the problem of cost. The noise reduction which will not ultimately pay for itself in increased customer acceptance is not generally warranted, since although the customer wants a quiet box he will generally not pay a premium cost to get it.

This paradox of attempting to get something for nothing is the challenge in noise control. Fortunately, logical reasoning processes in the design and development stages, coupled with available acoustical materials, keep noise within acceptable limits.

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Standards evolve to meet needs

A. T. BOGGS, III
ASHRAE Technical Secretary

Evolution of a Society standard relates to the need for such standard by the industry. Of the presently-active standards, a primary example of one that has followed the needs of the industry is ASHRAE Standard 16-56R, Method of Testing for Rating Room Air Conditioners.

In the early 1930's a joint committee sponsored by ASRE and made up of representatives from Nema, RMA, ACMA, and ASHVE compiled the initial standard Circular #13, Standard Method of Rating and Testing Air Conditioning Equipment. By 1939, sufficient progress had been made in air conditioning as to indicate a new circular for rating and testing self-contained air conditioning equipment used for comfort cooling. This standard was developed by a joint committee consisting of representatives of the same groups as for its predecessor.

During World War II, because of the need for military applications of refrigeration equipment, considerable advances were made in technology. The interest in air conditioning for residential and commercial applications grew tremendously following World War II. This was followed by a similar growth in production of units by a number of manufacturers. Although the bulk of the production was handled by the so-called older manufacturers, many new firms were started because of this increased market. Again, there was presented the need for an updated rating and testing standard. In 1949 the predecessor of the present ASHRAE Standard 16 was revised to cover the broad area of rating and testing of air conditioners. Earlier standards included not only the methods of rating and testing, instrumentation and apparatus, but also such subjects as flow measurement, cooling, heating and humidifying capacity tests.

Revisions, prepared in 1953 and 1956, paced the increased use of air-conditioning equipment for year-round applications. The inevitable, growing complexity of the equipment demanded new standards for the testing and rating of each type. At a national meeting of ASRE in 1956, the Standards Committee decided that the time had arrived when the standard on air conditioners should be revised so that separate types of equipment could be covered by individual standards. From this initial decision developed four new standards now designated as: ASHRAE 16-56R—Room Air Conditioners; 37P—Unitary Air-Conditioning Equipment; 39P—Unitary Heat Pumps; and 40P—Unitary Heat-Operated Air Conditioners. The proposed Standard 37P on unitary equipment is being reviewed now by the Standards Committee prior to submission to the Board of Directors. Project committees are developing actively Proposed Standards 39P and 40P. The revision of the basic standard covering room air conditioners, now designated as 16-56R, has been completed by the project committee and is being reviewed by the Standards Committee. It is anticipated that this will be available for Society review prior to the Dallas Meeting.

There are a number of principal changes being proposed for the room air-conditioner standard. It does not cover units using water-cooled condensers, as this type of equipment constitutes but a small percentage of the overall production of room air conditioners, and there is a trend to reduced production.

The heating capacities of room air conditioners are not included in the revised standard; this characteristic of certain types of room air conditioners probably will be covered by another standard. The "heat pump" window units would be tested by this new standard for cooling capacities.

One of the principal changes in the actual test apparatus specified in the standard concerns the use of a pressure-equalizing device in the wall between the room side and the outdoor side compartments of the calorimeter. This device would consist of one or more nozzles with the proper discharge chamber, exhaust fan, and manometers for measuring the air flow between the two rooms. It is felt that this would be better than the flexible flap indicated in the original standard.

There is a new provision for measuring the inlet air temperature to the condenser coil at the inlet louvres of the air conditioners instead of at the outlet of the reconditioning equipment. This method of measurement may overcome errors that were possible under the previous standard.

ASTM—ASTM Standards on coated and uncoated iron and steel sheet and strip have been compiled and are now available from ASTM at a price of $3. This is the initial issue of this compilation and includes 21 standards. Eleven of the standards are specified for steel sheet and strip, three for metallic-coated steel sheet, two for wrought iron sheet, and one specification for metallic coating materials. There are also 4 methods of test. This publication presents in a convenient form ASTM standards relating to sheet and strip steel materials both metallic coated and plain.

ARI—A recent release from ARI indicates that a certification program for unitary heat pumps has been initiated and it is hoped that the program may become effective some time in 1960. This certification program for heat pumps will parallel the already operative certification program for unitary air conditioners. When inaugurated, the heat pump program will provide for testing and rating by manufacturers and reporting ratings to ARI, for publication of ratings in an official ARI Directory, for random testing of units on the market by an independent testing laboratory under contract to ARI, and, for special tests on units on which complaints have been lodged. It has also been announced that the final 1959 issue of the Directory of Certified Unitary Air-Conditioners has been prepared and copies are available from ARI headquarters in Washington, D. C.

National Academy of Sciences—Of probable interest to engineers using industry standards will be the new directory "Industrial Research Laboratories of the United States" now being compiled by the National Academy of Sciences, National Research Council. The eleventh edition of this directory will be published in mid-1960. A publicity release indicates that approximately 6000 scientific and technological laboratories will be described in the new volume. For those not familiar with this publication, the directory has been published periodically by the Research Council since 1921. It is designed to provide government and private organizations engaged in research and developed with a convenient catalog of the national resources in industrial laboratory facilities.

Radiant heat control

in industrial plants

In order to provide comfortable working conditions in large industrial plants, serious obstacles must be contended with—heat released from the equipment, shape of buildings, and a lack of understanding of basic causes of discomfort. Often, the importance of radiated, radiant or infrared heat is not realized fully. Herein, radiant heat problems are described and solutions to them offered. At the same time, the significance of proper ventilation is stressed.

W. G. HAZARD

Many industries face severe obstacles, today more than ever before, in providing working conditions that are comfortable from the standpoint of heat. In heavier manufacturing more and more heat is released within the structure from equipment: machine tools use more power, motors are larger, faster heating cycles bring higher output and greater tonnages are passed through furnaces, ovens and driers.

Furthermore, the basic change in shape of buildings has accented the problem: they are no longer long and narrow with ample cross drafts; courtyards for ventilation are out of date; buildings are single-story and sprawling so most people in them are far from any outside window; vast expanses of glass walls invite solar heat in excess of what is good for comfort. For example, 6 sq ft of glass wall admits as solar heat the equivalent of a pound of steam per hr.

Finally, the shortcomings of many ventilating systems in certain types of plants stem from a basic misunderstanding. Plant supervisors and even some engineers are confused over what factors make people uncomfortable. Chiefly, this is because the importance of radiated, radiant or infrared heat is not appreciated.

W. G. Hazard is with the Industrial Relations Div, Owens-Illinois. This is a somewhat condensed version of a paper presented at the Industrial Ventilation Conference at the ASHRAE annual meeting at Lake Placid, N. Y., June 22-24, 1959 The full text will appear in the Symposium Reprint covering all papers presented at this Conference.

To clarify the topic, many useful forms of radiant heat, such as panel heating, infrared burners, and the like, with which heating experts are familiar will not be discussed. We are to deal only with the reduction of excessive thermal conditions that cause discomfort in industrial operations.

COMFORT – AHVR FACTORS

Four factors of the thermal environment affect man's comfort:

Air temperature
Relative humidity
Velocity of air blowing on him
Radiant heat, that is, the mean temperature and emissivity of surfaces and objects that surround him.

Additional factors, such as type of clothing, rate of work or metabolism, age, acclimatization, preexisting disease or physical impairments, etc., also affect the feeling of comfort; but these are nonenvironmental, and will not be included in this discussion.

The four AHVR factors can be measured with simple instruments: air temperature – with an ordinary thermometer, which must be shielded if there is much radiant heat present or the reading will be inaccurate; relative humidity – by a wet-bulb thermometer (sling psychrometer), plus a relative-humidity slide rule or set of tables; air movement – by a wind gage, which can take any of several forms, such as a swinging vane anemometer, a heated thermometer anemometer, thermistor, thermocouple, etc.; and mean radiant temperature—by a globe thermometer.

SAMPLE PLANT DATA

Some actual readings taken where men work in the vicinity of furnace operations may emphasize the magnitude of exposure to radiant heat. In one unusually hot plant a series of 17 tests were made during a period when the outside air temperature varied 2 F, from a low of 87 to a high of 89 F. The median workroom air temperature was 116 F (low 106, high 135, depending on location of test), or 28 F above outdoor temperature. The median globe thermometer temperature in the workroom was 138 F (low 117, high 155). $\bar{t}_g$ was thus 22 F above $\bar{t}_a$, where the bar over the symbol (–) represents the median value. Dewpoint inside during two-thirds of the readings was 69 F, and during the other third, 74 F. Median air velocity was 175 fpm (15 tests ranged from 125 to 300 fpm, while the other two were 1200 fpm). The median value for mrt (mean radiant temperature of surroundings) by calculation was 180 F (low 129, high 212).

Assuming a metabolic rate of 800 Btu/hr (standing, moderate work at machine, some walking about), the median heat load was made up as indicated in Table I.

TABLE I

Heat source	BEFORE shielding: Total heat load Btu/h	BEFORE shielding: Total heat load Percent	BEFORE shielding: External only Percent	AFTER shielding: Total heat load Btu/h	AFTER shielding: Total heat load Percent	AFTER shielding: External only Percent
M (metabolism)	800	26	—	800	39	—
C (transferred by convection)	550	18	24	550	27	45
R (transferred by radiation)	1700	56	76	680	34	55
Total	3050	100	100	2030	100	100

These figures illustrate what has been found repeatedly: in summer, in environments with a high radiant heat load, infrared or heat transferred by radiation often make up three-quarters of the heat falling on the person working there.

The efficacy of infrared shielding can be demonstrated easily. Follow-up tests made where reflective shielding has been judiciously installed show a marked reduction in radiant heat. Naturally, all the radiant heat load cannot be eliminated where there are multiple hot sources; but it is not difficult to get a 60 per cent reduction in R. The improvement is shown in the three right-hand columns of Table I. The heat load is now fairly evenly divided between M, C and R. An attack to lower C, through proper air supply systems, evaporatively cooled air, etc., will give good returns. But to attempt to reduce the original radiant load by ventilation instead of shielding would have been financial waste, poor engineering and even an actual impossibility.

Many such examples of what a direct attack on radiant heat will accomplish are available; and all improvements, even though less extreme than the above, are received by people working there with gratitude. An example is a pulpit that straddles a conveyor between the soaking pits and rolls of a 54-in. mill. Aluminum was installed under the pulpit floor and up the side walls to observation windows 3 ft above floor level. Inside, the surface temperature of the roof, which received no primary radiant heat was 95 to 105 F, whereas the floor was a trifle cooler, 95 to 100 F. The floor was bathed in heat from the conveyor below, but the aluminum sheathing warded it off. Further, the shielded wall below the windows was 85 to 90 F, while the unprotected wall above the same windows (slightly farther from the source) was 140 F. By calculation the unshielded wall was radiating heat to the occupants at 56 Btu/hr/sq ft; whereas, if all the walls had been at the temperature of the shielded wall, the occupants would radiate heat to the pulpit walls at a rate of 8 Btu/hr/sq ft. These pulpits are sometimes air-conditioned. But not many can afford to install air conditioning without first trying radiant heat shielding.

Another example is a brick furnace stack, square in cross section. At an operator's platform next to the stack, its side was shielded with a piece of commercial corrugated siding mounted about 2 in. away from the brick. Temperature readings were made as shown in Table II.

TABLE II

.... Surface	Globe thermometer F	Air temperature F	Globe above air F
Side of brick stack:			
Unshielded	161	93	68
Shielded	99	89	10
Reduction due to shielding	62	4	—

Air temperatures were only 4 F apart, at the unshielded compared with shielded walls of the stack, while the globe readings differed by 62 F—thus establishing the basic point: that the problem is one of radiant heat, not convection. A simple sheet of aluminum lowered the globe reading by some 62 F, or about 40 per cent, down to within 10 F of the air temperature.

CONTROL OF RADIANT HEAT

After determining through globe thermometer readings that a radiant heat problem exists, the solution comes from elementary physics, and this solution is not ventilation. Control is accomplished as follows:

At the source—Amount of heat radiated depends on (a) temperature of the source, and (b) emissivity of the source.

The surface temperature of the source sometimes can be lowered, as by applying thermal insulation to a hot tank, oven or furnace wall. Often the source temperature cannot be tampered with because it is inherent in the process. Forming of glass and metals requires them to be in a molten state, and they frequently radiate huge quantities of heat to men working nearby.

The source emissivity can sometimes be reduced, with a consequent reduction in energy radiated. Thus aluminum pipes or ducts carrying hot liquids or gases radiate much less heat than black iron systems. The same is true of aluminum or polished stainless steel vats and containers that are hot. High temperature aluminum paint, whose emissivity may be 30-35 per cent, which is clearly better than 90-95 per cent for ordinary paint or black iron (but much above sheet aluminum's low emissivity of 5-10 per cent), may bring considerable improvement.

Shields—When the source itself cannot be altered, infrared-opaque shields can be interposed between it and the men. Just as in sunlight, they set up shadows, in this case invisible infrared shadows.

Nearly every solid matter, even glass, is virtually opaque to radiant heat, but the choice of material, nevertheless, is important. The shield should be such that it does not heat up and become a troublesome secondary source of radiation. There are two possibilities: shields that are intentionally heat absorbing but have a cooling system, usually water; and reflective shields. Sheet metal or heavy mats over which water flows constantly are used, as are hollow or envelope shields, or plate coils, through which cooling water passes.

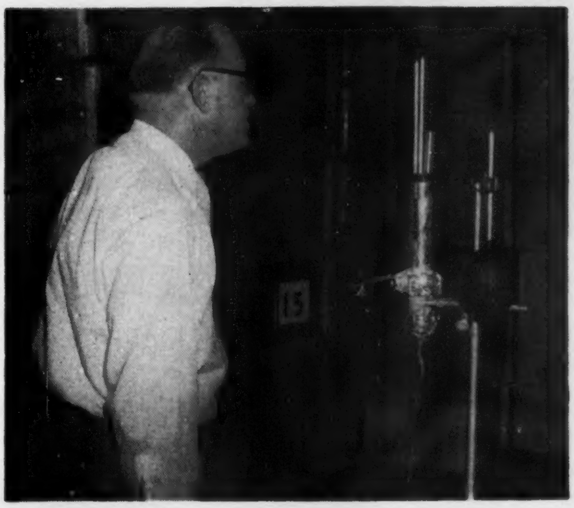
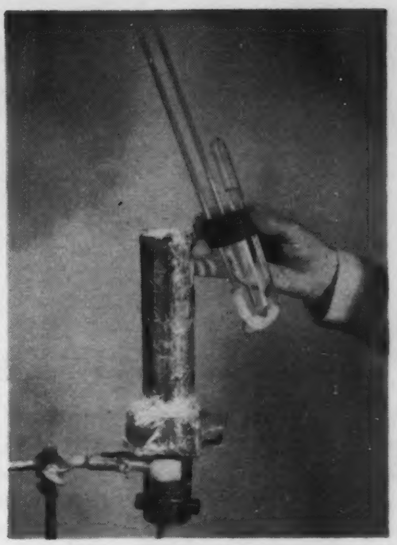


Fig. 1 Radiant heat is best measured by a shielded, aspirated wet and dry bulb thermometer here shown as to component parts and in use



For low cost and ease of construction, reflective shields are finding wide use. Most shiny or highly polished metals have the property of reflecting infrared energy. Aluminum is preferred, since in its commercial grades it will reflect 90 per cent or more of the incident infrared. It retains its good reflectivity even when dry dust settles on it or after it has become oxidized. It deteriorates as a reflector when it becomes moderately coated with oil, or with any other material that interferes with the aluminum-to-air surface. If it becomes blackened, or is painted with ordinary paint, it becomes just another black body.

However, heat shields are not a cureall. To be effective several points must be considered:

1 A fundamental law of physics is that poor absorbers are also poor emitters. Sheet aluminum follows this law as long as both sides have uncoated metal-to-air surfaces. If one side is imbedded in, or laminated to, or coated with some other material, the overall effectiveness of the shield is lessened.

2 The shield should be mounted a couple of inches out from the hot surface, and physical contact with the surface should be avoided. In this way an air channel is formed so the shield is cooled by convection; and also, it does not heat up through direct conduction.

3 Occasionally, the reflection of heat from the shield, which is the reason for using it, may increase the load on the incident (hot) side enough to unbalance operating conditions there or overheat machinery.

4 Never install shields in locations where they will interfere with natural or forced ventilation, or mancooling systems.

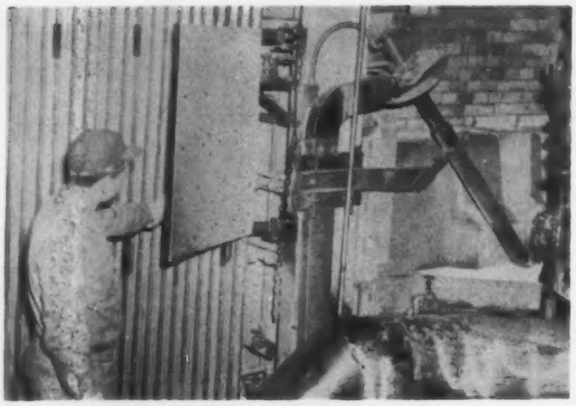
5 On some jobs shields may handicap the operator by restricting his field of vision, in which case they should be adjustable by sliding, hinging, or removing. It then becomes highly important that the men understand what the shields should accomplish and how they work. To solve heat problems by setting up invisible shadows, instead of getting a fan to blow more air around, is not an obvious answer to most people. Individual or group explanation of the elements of radiant heat should be given, always accompanied by some simple demonstration.

Fig. 2 Aluminum shielding reduces radiant heat flow to adjacent areas with high effectiveness

6 As a corollary, adjustable shields are often removed in winter. A place is needed to clean and store them. Otherwise, the pieces of aluminum find their way to the scrap or salvage pile. When spring rolls around, a hurry-up call goes to the maintenance department to rebuild all the shields. Result is needless cost and inevitable irritation—especially when this occurs the second year. A program for maintaining both fixed and movable shields is essential.

7 While our concern here basically is to make people more comfortable, shields can often be used to protect equipment from overheating—motors, controllers, etc., and to lower the load on air-conditioning systems. The shield on the stack mentioned above protected successfully a large motor that previously had burned out twice.

Opaque screens sometimes cannot be used because the process has to be continuously observed. Plate glass with an infrared reflective coating is commercially available which can be used for windows. It may be either a borosilicate (low temperature expansion coefficient) or heat-treated glass, to minimize breakage. Commercial heat-absorbent glass is not as suitable, since it becomes hot and is a source of secondary radiation.



A

B

C

Fig. 3 Shielding may be of several forms: A — Hinged reflective shield at pot burner; B—Asbestos-faced corrugated sheeting; C — Formed aluminum sheeting offers high reflectivity for this stack

Fig. 4 Some sort of more ultimate protection for the individual? An infra-red reflective helmet and jacket

A low roof made hot by solar radiation can be regarded as a type of shield for the protection of people below. Their heat load can be reduced by insulating the roof, making it reflective with a light-colored surface (appropriate because solar heat contains much short-wavelength radiation), or by cooling it through evaporation, which means a water spray or sprinkling system.

Personal protection — In some radiant heat exposures the only answer is to protect the men individually.

Occasionally an open-door booth or shelter can be established in the work area to which a machine operator can retire at intervals, when the job is running smoothly. It should be sheathed in aluminum. A mancooling air duct and adjustable diffuser should be provided in the booth.

In more severe situations, when the exposure is intermittent, as for making adjustments or emergency repairs, reflective clothing (helmet, gloves, jacket, trousers, or coverall) is the answer. The clothing may be of any type—cotton, asbestos, nylon, glass—with its outer surface made shiny by aluminum especially applied. When contact with a hot surface, like a furnace wall, is likely, the garment should be made of quilted material, as with a fluffy fiberglass liner, for example. This provides conduction insulation, while the shiny surface reflects radiation. So effective are these garments that many jobs are being done today which were formerly considered unbearable. Two drawbacks will be recognized: the garments act to some degree as vapor barriers so the body cannot breathe freely through them, and the shiny surface is not completely durable. The latter deficiency is being steadily worked on, and improvements in wear and abrasion resistance have been made recently. The former defect can sometimes be helped by using partial protectors like aprons, gloves with aluminized backs only, shoulder shields, and the like, that do not encase the body completely. Perhaps aluminized cloth with perforations cut in it would be a partial answer.

Finally, an aluminized hand cream has been marketed, which can be spread directly on exposed skin, or coated on cotton work gloves, to make them heat reflective.

Heat shields, designed and installed after close study of the problem, are a first line of attack on radiant heat — and incidentally, an inexpensive weapon in view of the good they can do—but they are not a substitute for proper ventilation. This subject is well covered by other papers. It is touched on here, first, to dispel any idea that radiant heat control is the sole solution to heat problems; and second, to re-emphasize the modern concept in ventilation for hot plants.

Gravity ventilators, or even power roof fans, by themselves, are no longer considered sufficient. A supply system needs to be included, or the roof fans will be starved for air. Even more important today is this idea, "How can the air that eventually goes out the roof best be delivered to the work spaces?"

The supply air becomes doubly effective when it is brought in from outdoors and delivered at the work stations where men get its

direct cooling effect. This becomes especially important in large area, single story plants. Supply air should be piped to all parts of the work space, and delivered through diffusers 8 or 10 ft above floor level. Diffuser design and outlet velocities need careful attention, and above all, the individual whom they serve should be able to adjust the direction of throw and air flow rate.

EVAPORATIVE COOLING

In excessively hot environments evaporative cooling of sections of the supply air system gives spot cooling and welcome additional relief. Cooling is by a non-refrigerating method, such as an air washer. No space cooling is involved; that is, no effort is made to reduce greatly the average temperature of the whole building. This method is not air conditioning in its popular meaning. It merely makes it possible to blow medium-temperature air from a mancooling diffuser to the person, instead of hot air. The air from the spot cooler may be only 10 to 12 F below the temperature of uncooled air – but when the thermometer gets around 100 F, such cooling is helpful.

Outdoor air in passing through the washer is cooled and becomes saturated. Its dry-bulb reading is within a degree or two of the wet-bulb. This is a conversion to latent heat, and heat is not extracted as in refrigerative cooling. The washed air can pick up 4 to 6 F when passing through the ductwork in hot areas, and still be delivered cool enough to do some good. To minimize this temperature pickup, the ducts should be insulated where room-air temperatures are high, or made of aluminum where much radiant heat is present. Duct velocities should be high, say, 3000 fpm, to speed up the time of transit through hot zones.

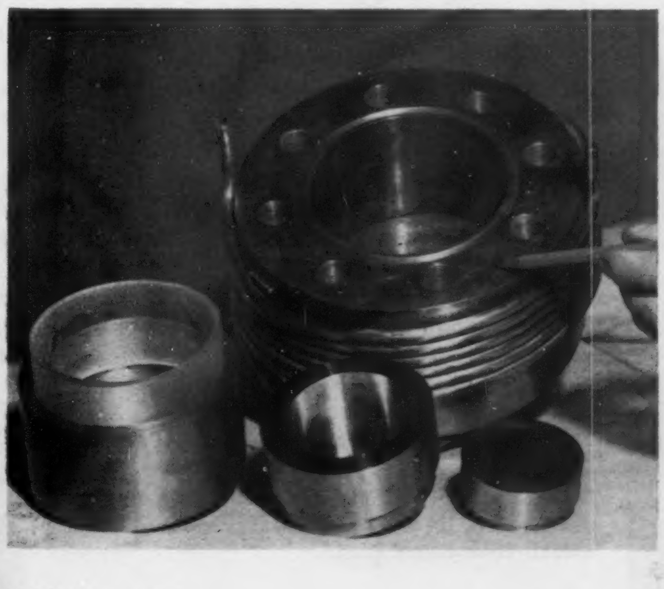
Four precautions to be kept in mind are:

1 Use all outdoor air and never recirculate in hot weather.
2 Be sure there is enough air motion around the men who are to be benefited by the spot coolers—300 fpm minimum, up to 1000 fpm. Individually controlled dampers, permitting 80 per cent velocity turndown, are essential for personal comfort, especially in off-seasons like spring and fall.
3 Use a split system, one part having evaporative cooling, the other bringing in raw outside air to the room. Provide roof exhaust fans. A two- to five-fold greater volume of unwashed air than washed prevents any possible accumulation of spent washed air, and also holds down the overall temperature level.
4 Do not use evaporative cooling for offices (except where this is acceptable practice as in the Southwest), or for areas where hot operations do not exist.

The significance of temperature of mancooling air is sometimes overlooked. Quite apart from radiant heat and its peculiarities, when a person is hot, he seeks a higher and higher velocity of mancooling air. Not infrequently, this can make him hotter. If the air from the mancooler is above his skin temperature, above 95 F, increasing its velocity will bring increasing amounts of heat to him by convection. It will also increase the evaporation rate of this sweat, and hence promote cooling, but not at as fast a rate, after a certain point is reached, as he is heated by convection. Blowing hot air on him thus can be highly undesirable. Hence the reasons for holding this air to below 95 F are physiologically sound, and often evaporative cooling with an air washer is the only economical way of accomplishing this.

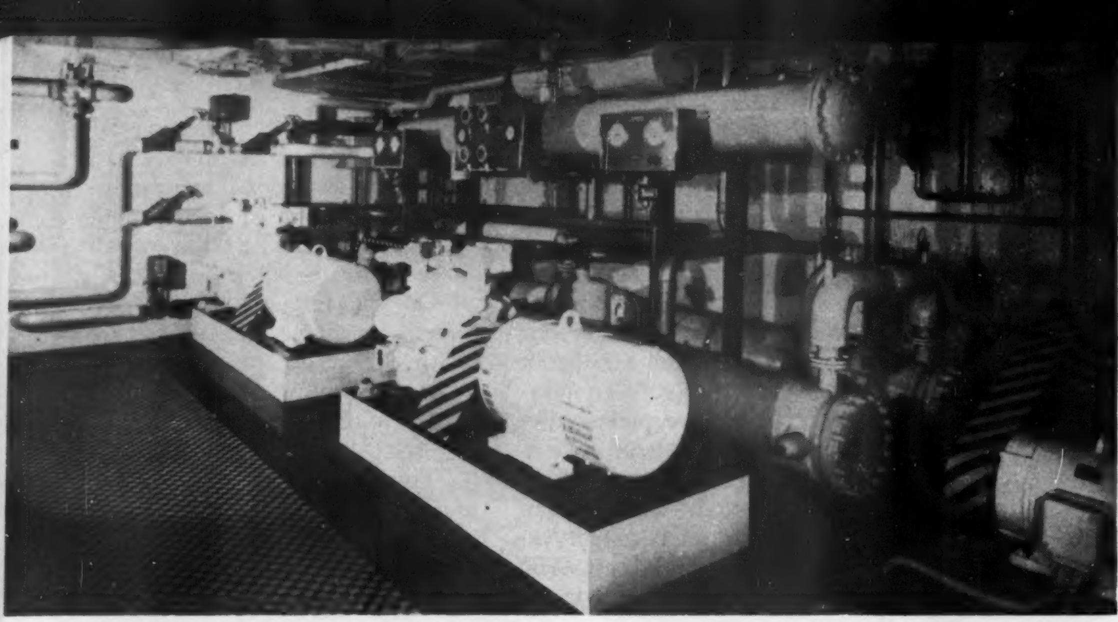
WHAT IS THERMAL CONDUCTIVITY OF HYDROGEN?

These pieces of machined beryllium copper alloy are the heart of a new thermal conductivity apparatus enabling the National Bureau of Standards Cryogenic Engineering Laboratory at Boulder, Colo., to learn more about liquid and gaseous hydrogen and what effect a change in pressure or temperature might have on the ability of heat to travel through these two forms of hydrogen. Copper is used in the apparatus for its heat conductivity and the beryllium gives it the strength of stainless steel, needed in order to withstand the pressures which will be applied.



To hold the lid on the pressure chamber, built to withstand pressures up to 4,500 psi, eight bolts of ¾-in. diam stainless steel are used. The parts in the foreground fit together as part of the chamber like a telescope.

A versatile unit was specially designed for this project because of the stringent demands that will be made on the apparatus not only in pressure but also temperature which will fluctuate from —450 F to room temperature. Temperature measurement requires the use of small platinum thermometers which are coils of platinum wire, wound in a coil around a core and fitted into a chamber.



Gleaming white upon stepped bases of black on a checkerboard floor, these two direct-connected, motor-compressor units are flanked by carefully placed air conditioning system components, piping and ducts, color-coded and arranged for visual appeal.

Double duty served

by Swiss dress shop air conditioning equipment

There are relatively few air conditioning systems in Basel, Switzerland, so when the Fashion House of Feldpausch went modern it did so in quite a spectacular way, as may be noted from the cover illustration on this issue.

Dramatizing the installation of 165 ton of cooling equipment, an elaborate setting was created for it. Visitors are welcome in a basement area where color-coded equipment, piping, ducts, instruments, enclosures and bases have been given an integrated engineering and architectural lift.

Of the 165 ton capacity, 135 ton are supplied by two Carrier compressors supplemented by pre-cooled well water which holds a temperature of 52 F. Compressors operate at 1450 rpm and are direct-connected to electric motors. The system uses Refrigerant-22.

The installation includes two water coolers of the Acme through-tube type which reduce the temperature of the water from 53 to 43 F. The cooling medium is then sent to three dehumidifying coils.

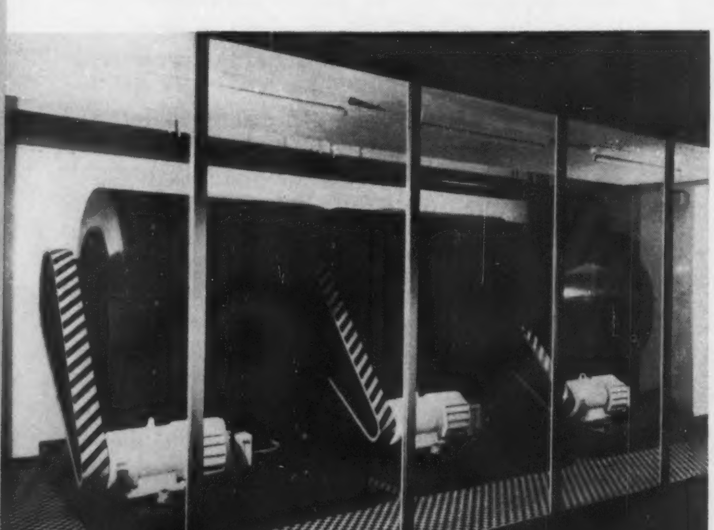
Air circulation and distribution is maintained by three fans with a capacity of approximately 60,000 cfm. As part of the dramatization of the equipment, one side of the fan chamber is glass enclosed and the units there have been color-coded maroon with white motors, a yellow and black drive enclosure and the same black and white checkerboard flooring that serves the rest of the installation.

Washer has an effective area of about 110 sq ft.

Because of a frequently expressed sensitivity to drafts on the part of many Europeans, the diffusion system for the Feldpausch store has been designed with unusual care. Through each of the store's six floors has been hung a perforated ceiling to insure the utmost in the uniform distribution of air.

As indicated previously, there are few air conditioned stores in this country and both a pride of ownership and a highly developed merchandising instinct have led management to make the most of inherent possibilities in this situation. Not only do visitors see the equipment under the most favorable circumstances, some of it behind glass, but the emphasis upon controlled climate seems to induce a more receptive attitude toward comfort in the purchase of the merchandise in this predominantly dress shop enterprise.

Again, a corollary purpose in the glamorizing of the machine room has been to make cleanliness, good housekeeping and proper maintenance a matter of pride and efficiency for the operating staff.



Behind severely plain glass paneling and fluorescent-lighted, these ventilating fans are part of an overall color-coded plan.

To control condensation

between panes of double windows

Discussing condensation between panes of double windows, the authors analyze the factors involved in the transfer of water vapor to and from the air space. They present facts on the pressure distributions across the panes with respect to the relative air resistance of the cracks around inner and outer panes, and consider the role of buoyancy forces in transferring vapor by air flow. Vapor transfer by diffusion is compared with transfer by air flow for a specific window, and these results are in turn compared with observations of window condensation in cold room installation. Buoyancy forces are found to be the most effective means of condensation control. The authors also discuss the effect such venting has on heat transmission.

A. GRANT WILSON
Member ASHRAE

E. NOWAK

That double windows are of value in reducing heat transmission through window areas and in permitting higher inside relative humidities during the winter, without excessive condensation on inside glass surfaces, is well known. Some form of double-window arrangement is used in most houses in regions having low winter temperatures.

There is an increasing use of double windows in commercial and industrial buildings, especially when year-round air conditioning is employed.

Condensation of water vapor between the panes, on the inside surface of the outer pane, is a common occurrence with most types of double windows except the factory-sealed variety. A small amount of such condensation is accepted generally as inevitable. When it begins to obstruct seriously the view through the window for long periods, however, or when the run-off contributes to the deterioration of surrounding materials, there is reason for concern.

Condensation will occur on the inside surface of the outer pane whenever the temperature of that surface at any point is below the dewpoint temperature of the air-vapor mixture in the space between the panes. This ultimately will occur, with outside temperatures lower than inside, if the gain in water vapor to the space is greater than the loss. An approach is developed here for determining under what conditions condensation will occur in a given window, or alternately, for designing a window to be free of condensation under given conditions.

Transfer by diffusion—Water vapor generally moves into and out of the air space of a double window under two forces. It moves through the materials and cracks in the window assembly by diffusion as a result of differences in partial pressure of the water vapor, and it is transferred as a component of the air which flows through the cracks in the assembly under total air pressure differences.

Most of the materials used in window construction are relatively impermeable to vapor, except wood and it can be made relatively resistant through painting. The cracks and openings in the assembly will thus usually provide the major paths for vapor flow by diffusion. The permeability coefficient for air water-vapor mixtures is independent of relative humidity but does vary somewhat with temperature.

A. Grant Wilson and E. Nowak are associated with the National Research Council of Canada, in the Div of Building Research. This is an abstract of "Condensation Between Panes of Double Windows" as presented at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959. A contribution of the Div of Building Research, this will be published in full, with the approval of the Director of the Div, in ASHRAE Transactions.

For purposes of this discussion, however, it can be assumed constant. The vapor flow by diffusion in to and out of the air space is then directly proportional to the pressure difference across inner and outer panes.

The vapor pressure differences across inner and outer panes are dependent on inside and outside temperatures and relative humidities. Table I illustrates the possible order of these pressure differences. The values have been calculated taking inside room and air-space vapor pressures corresponding to saturation at the temperatures of the inner surfaces of the inner and outer panes respectively. Outside vapor pressures were taken to correspond to saturation at the outside air temperature. Surface temperatures were calculated for an inside air temperature of 70 F assuming an over-all "U" value for the window of 0.53 and inside and outside surface conductances of 1.5 and 6.6 respectively (based on heat transmission values for windows in the 1958 ASHAE Guide).

The vapor pressure difference across the inner pane is several times greater than the vapor pressure difference across the outer pane in all cases; the ratio of these pressure differences increasing with decreasing outside temperature. Thus, to maintain outflow equal to inflow, the effective resistance of the inner glazing to vapor flow by diffusion must be many times that of the outer glazing.

Pressure differences — Several factors may contribute to total pressure differences across windows. Air movement around a structure from wind action will result in a pattern of pressures that depends on many factors. It can be assumed in the simplest cases that the pressures on leeward walls are below barometric while those on windward walls are above it. Pressures inside the building resulting from wind action will depend on the distribution of air leakage throughout the building enclosure.

If the leakage characteristics of all exposures are similar, the pressures inside will probably be less than barometric. As a basis for discussion, pressures on the windward side, leeward side, and inside of 0.8, minus 0.5 and minus 0.2 velocity heads respectively can be assumed. This leads to total pressure differences of 1.0 velocity head across windward windows and 0.3 velocity head across leeward windows.

TABLE I

RELATIVE VAPOR PRESSURE DIFFERENCES ACROSS INNER AND OUTER PANES

Outside Temperature F	Ratio of vapor pressure across inner pane to that across outer pane
40	7
20	9
0	18
—20	29
—40	51

For purposes of heat loss calculations, the flow from outside to inside resulting from the higher pressure difference is significant. It is the flow from inside to outside occurring under the lower total pressure difference, however, that is significant in connection with condensation between double windows, since this results in a net gain in water vapor to the entrapped air space.

A second major factor contributing to total pressure differences across windows is chimney action between air in a building and the outside induced by temperature differences. As a result, air tends to flow into the building through lower openings and leave through upper openings. With no other forces acting, there is a neutral zone somewhere between at which inside and outside pressures are equal. The pressure difference across the walls of any enclosed space due to chimney action can be calculated easily providing the temperature differences and the level of the neutral zone are known. Unfortunately, information on neutral zone locations for both residential and commercial buildings is extremely limited.

Available experimental records for residences[1] suggest that the actual levels are considerably higher than would be predicted on the basis of the vertical distribution of window and door cracks. In single-story houses, the neutral zone may be above first floor windows, while in two-story houses it may be at the level of second story windows. Recently published information on pressure differences across entrances of tall buildings suggests that the neutral zone location in such structures may be well above mid-height.[2]

Since flow from the building to outside occurs only above the neutral zone, the level of the neutral zone is significant in connection with condensation between the panes, windows above the neutral zone being much more likely to have such condensation.

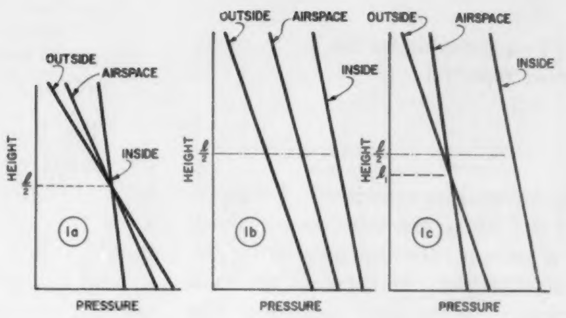
Pressure distributions — In double windows the resistance to air flow of the passages leading from inside the building to the air space may be different from that of the passages leading from the air space to outside the building. This is not significant with respect to heat loss, where over-all leakage values are important. The relative air leakage characteristics of passages into and out of the air space may be highly significant, however, with respect to condensation between panes. This is apparent when the distribution of total pressures and resulting air flows across double windows are considered.

Fig. 1 illustrates the variations in pressure with height across a double window, subjected to a temperature gradient. In representing the relative slopes of the pressure distribution curves, it was assumed that the temperature of the air space was closer to outside temperature than to that inside. The pressure distributions shown in Fig. 1(a) where the neutral zone occurs at mid-height will result whenever the resistance to flow above and below the neutral zone, around either or both panes, is equal. For example, this will occur with the passages uniformly distributed around the periphery of one or both panes.

Under these circumstances the air-space pressure is between inside and outside pressures at all levels. Thus, inside pressure is higher than air-space pressure above the neutral zone and lower below the neutral zone. Similarly, the air-space pressure is higher than the outside pressure above the neutral zone and lower below the neutral zone.

The air flow resulting from these pressure distributions will depend on the locations and resistances of the passages. Flow will

Fig. 1 Variations in pressure with height across a double window



vary from zero through passages located at the neutral zone to a maximum through passages at the top and bottom of the window. The air space can interchange air with both the inside and outside; the relative amounts depend on the relative resistance to flow of the passages. If the inside of the window were completely sealed the air space would interchange air with the outside only.

Figs. 1(b) and 1(c) illustrate the variations in pressure with height across a double window subjected to a temperature gradient where the neutral zone is below the window, i.e., where the total pressure inside is greater than the total pressure outside at all levels.

Fig. 1(b) represents the pressure conditions when the resistances to flow around both panes are equal. The air-space pressure is approximately mid-way between inside and outside pressures and flow is from inside to out through all openings. The air space does not interchange air with the inside or outside. Fig. 1(c) represents the pressure distribution when the resistance to flow through passages of the inner pane is much greater than that through passages of the outer pane, these being evenly distributed above and below the mid-height of the window, with equal openings top and bottom. Under these conditions, the pressure at the top of the air space is greater than outside and the pressure at the bottom is less. The air space can interchange air with the outside, all outflow passing through the upper passages of the outer pane. This consists of both the inflow through the lower openings of the outer pane and the flow from inside to the air space through passages of the inner pane.

These conditions can only be approached if the pressure difference required to move all the air through the upper openings in the outer pane is less than that created by the difference in weight between the columns of air in the space between the panes and the outside.

Air flow rates—The foregoing discussion of pressure distributions across double windows has shown that, with the air-flow resistance of the inside pane sufficiently higher than that of the outside pane, and with the openings in the outside pane located at the top and bottom of the window, the air space can interchange air with the outside even when an over-all flow from inside to outside occurs. The relationship between these air flows and condensation between the panes can be shown by a simple mass balance.

If vapor flow by diffusion is neglected, the net gain of water vapor by the air space as a result of air flow from the inside is approximately equal to

$$Q_i\, d_i\, (W_i - W_s) \qquad (1)$$

where

Q_i = vol. rate of flow into the air space from the inside
d_i = density, inside air
W_i = humidity ratio, inside air
W_s = humidity ratio, air in space between panes

TABLE II

MINIMUM RATIOS OF OUTSIDE TO INSIDE AIR FOR SPECIFIED CONDITIONS

Outside Temperature F	Ratio of Outside Air to Inside Air Q_o/Q_i
40	7
20	9
0	16
—20	25
—40	41

The net loss of water vapor by the air space as a result of air flow from the outside is approximately equal to:

$$Q_o\, d_o\, (W_s - W_o) \qquad (2)$$

where

Q_o = vol. rate of flow into the air space from the outside
d_o = density, outside air
W_o = humidity ratio, outside air
W_s = humidity ratio, air in space between the panes

To prevent condensation the net gain in water vapor by the air space must equal the net loss of water vapor by the air space.

Equating Equation (1) to (2) it can be shown that,

$$R = \frac{Q_o}{Q_i} = \frac{d_i}{d_o}\,\frac{(W_i - W_s)}{(W_s - W_o)} = \frac{T_o\,(W_i - W_s)}{T_i\,(W_s - W_o)} \qquad (3)$$

where

R = minimum ratio of outside to inside air to prevent condensation between panes
T_o = Abs. temp. outside air
T_i = Abs. temp. inside air.

The values of the minimum ratio of outside to inside air to prevent condensation at different outside temperatures will depend on inside temperatures and relative humidities. Values given in Table II are based on humidity ratios for inside, air space, and outside corresponding to saturation at the temperature of the inner and outer panes and of outside air respectively, using the same conductances as for Table I. The value of the minimum ratio increases with decreasing outside temperature.

Test window—Laboratory studies were carried out, based on the foregoing considerations, to determine the degree of venting of the air space to outside required to overcome condensation between the panes of a simple wood sash window. Details of the unit are given in Fig. 2. The inner glazing, contained in a metal frame with a rubber gasket, was fixed to the wood sash with aluminum clips while the outer glazing was sealed to the sash with glazing compound. The resistance to air and water-vapor flow around the outer pane was altered by opening or closing ¼-in. diam vent holes provided in the

top and bottom members of the sash. The vent holes were approximately 1½ in. long, measured along their center-lines.

Measurements — In discussing the mechanisms of vapor transmission into and out of the air space of double windows, transfer by diffusion and by air flow have been dealt with separately. Both mechanisms will operate simultaneously, however, and will interact in an actual installation, making it impossible to determine their relative contribution to vapor flow. Separate measurements were therefore made of the water vapor diffusion and air flow characteristics of the cracks around the inner pane and of the vent holes around the outer pane as a means of assessing the relative importance of the two mechanisms.

Fig. 3 shows the arrangement used for measuring the vapor diffusion characteristics of the cracks around the inner pane. There were no vent holes around the outer pane in the sash used for this study and the wood was sealed with wax to make it essentially impermeable.

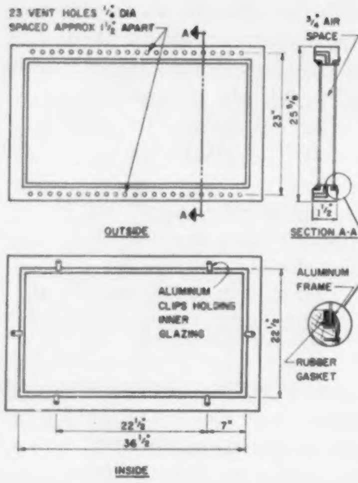


Fig. 2 Test assembly for the experiment reported

The procedure consisted of supplying dry air to the window air space at a known rate and measuring the humidity rise. If there is no total pressure difference between the space and outside, vapor transfer is by diffusion only. At equilibrium, the amount of vapor transferred by diffusion through the cracks around the inner glazing is equal to the gain in moisture of the air flowing through the window space. Thus,

$$M = D\,\Delta p_v = Q\,d_a\,(W_1 - W_2) \tag{4}$$

where

M = rate of water vapor transfer past the inner glazing, grains per hr

D = vapor diffusion coefficient for the inner glazing, grains per (hr) (in. Hg)

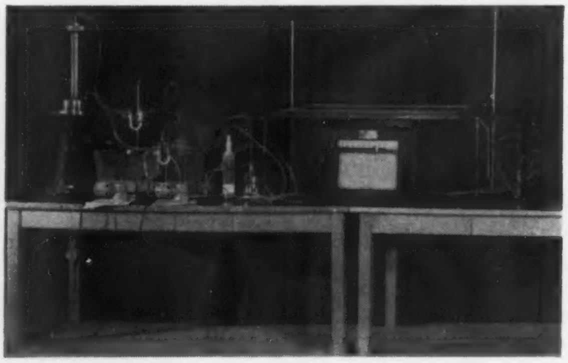
Δp_v = difference in partial pressure of water vapor across the inner glazing, in. Hg

Q = vol. rate of flow of dry air being circulated, cu ft per hr

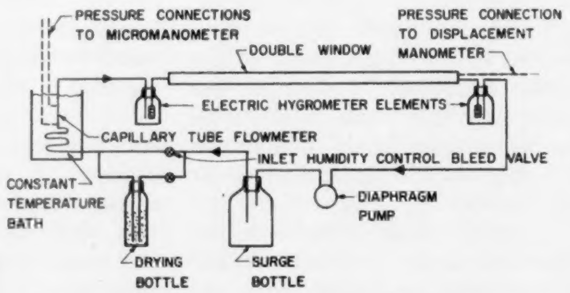
d_a = density of air being circulated, lb per cu ft

W_1 = humidity ratio of circulating air at inlet to the window space, grains per lb dry air

W_2 = humidity ratio of circulating air at the outlet from the window space, grains per lb dry air.



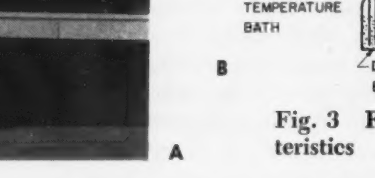
A



B

Fig. 3 For measuring vapor diffusion characteristics

Fig. 4 Permeability cup and copper tube



The humidity of the air being circulated was measured at the inlet and outlet of the window space with two electric hygrometer elements. These were calibrated prior to and following the test in an apparatus providing controlled conditions with an absolute accuracy of ±0.1% relative humidity. The conditions in the room in which the tests were carried out were controlled at 72 F and 50% relative humidity within close tolerances.

The air was circulated by a small diaphragm pump and the rate was measured with a capillary tube flowmeter in conjunction with a sensitive micromanometer. A displacement-type micromanometer in conjunction with a single pan analytical balance was used throughout the test to measure the total pressure difference between the window space and the laboratory. A change in the balance reading of 1 milligram corresponds to about 0.0001 in. of water. Since there was no change, it can be concluded that the total pressure difference across the inner glazing was essentially zero throughout the test.

The vapor diffusion coefficient, based on Equation (4), using the average of initial and final calibrations for the electric humidity sensing elements was 1.89 grains per (hr) (in. Hg). The possible error in the measurement is about ±20%.

Rather than attempt to make direct measurements of the vapor diffusion coefficients of the ¼-in. diam vent holes, tests were carried

out on a ¼-in. diam copper tube with 90° bend, mounted in a permeability cup as shown in Fig. 4. The tests were carried out in a conditioned cabinet maintained at 73 F and 50% relative humidity. The open end of the tube was protected from direct impingement of air being circulated within the cabinet. The permeance obtained for the tube as shown in Fig. 4 was 90 grains per (hr)(sq ft)(in. Hg).

Air flow tests were carried out on the inner glazing to determine its leakage characteristics during the vapor transmission measurements. The method consisted of supplying air to the window air space at the rate required to maintain a given pressure difference across the window. Pressure differences greater than 0.005 in. of water were measured with a micromanometer sensitive to about 0.001 in. of water; pressure differences less than this were measured with the displacement manometer and analytical balance sensitive to 0.0001 in. of water. Subsequent calibration of this manometer has shown that the error in measurement at pressures less than 0.005 is about plus 15%.

Test air flows less than about 2 cu ft per hr were made with the capillary tube flowmeter. At higher flows a calibrated variable area flowmeter was used.

The results of the air flow measurements on the inner glazing are shown in Fig. 5. The density of the air during measurements was essentially 0.075 lb per cu ft. The air flow was found to be independent of the direction of flow at pressures less than 0.015 in. of water.

Measurements were made to determine the air flow characteristics of the ¼-in. diam vent holes in the sash around the outer pane, using the equipment referred to above. The flow through the 46 vent holes was not exactly 46 times that through the single vent hole since the holes varied in resistance, having been drilled by hand. Fig. 6 shows the flow characteristics of 46 vent holes in the range of pressure differences normally provided by chimney action between the air space and outside. All flows have been converted with reference to air at 0.075 lb per cu ft.

Total pressure – To assess the relative importance of vapor flow by diffusion and by air flow in relation to condensation between the panes of double windows, it is necessary to consider what total pressure differences are available across the inner and outer panes at different conditions of over-all inside-outside pressure difference. This can be done with reference to the pressure distributions illustrated in Figs. 1(a) and 1(c).

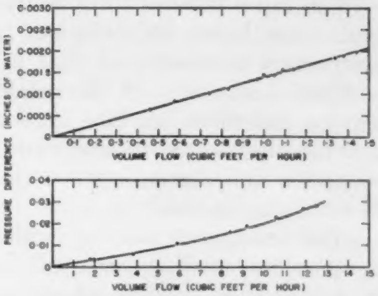


Fig. 5 Results of air flow measurements on the inner glazing

With the center of the window at the neutral zone, as illustrated in Fig. 1(a), the pressure difference across the bottom and top openings of the outer pane is,

$$p_o = 1/2\ (d_o - d_s)\ g/g_c \qquad (5)$$

where

p_o = pressure difference across top or bottom of outer pane, lb per sq ft

l = height of window space, ft

d_o = density of outside air, lb per cu ft

d_s = density of air in window space, lb per cu ft

g = acceleration of gravity, 32.2 ft per sec²

g_c = proportionality constant, 32.2 lb mass ft per lb force sec²

This can be expressed as

$$h_o = 3.82\,l\,\frac{T_s - T_o}{T_s \times T_o} \qquad (6)$$

where

h_o = pressure difference across top or bottom of outer pane with window at neutral zone, in. of water

T_s = mean temperature of air space, °R

T_o = outside air temperature, °R

Similarly the pressure differences across the inner pane at the top or bottom with the window at the neutral zone is,

$$h_i = 0.00721\,l\,\frac{T_i - T_s}{T_s} \qquad (7)$$

where

h_i = pressure difference across top or bottom of inner pane, in. of water

T_i = inside air temperature, °R

With the center of the window above the neutral zone as illustrated in Fig. 1(c), the window air space pressure depends on the relative resistance to air flow of cracks or openings around the inner and outer panes. Flow from outside to the air space occurs through the openings around the outside pane below level l_1. This vol., together with the air flow through cracks around the inner pane, flows through openings around the outer pane above level l_1. If the ratio of outside air to inside air is that defined by Equation (3) then,

$$Q_T = Q_o + \frac{Q_o}{R} \qquad (8)$$

where

Q_T = flow through cracks around outer pane above level l_1

But Q_T and Q_O can be expressed in terms of measured flow characteristics. For the test window the relationship is simple since, as shown in Fig. 6, flow is directly proportional to pressure difference at the small pressure differences produced by chimney action and the vent holes are located only at top and bottom. Thus,

$$Q_o = C_o\ (d_o - d_s)\ l_1\ g/g_c \qquad (9)$$

and

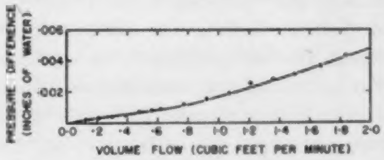
$$Q_T = C_o\ (d_o - d_s)\ (l - l_1)\ g/g_c \qquad (10)$$

where

C_o = flow coefficient for vent holes

l_1 = distance from the bottom vent holes to the level where air space and outside pressures are equal, ft

Fig. 6 Flow characteristics of 46 vent holes in the normal range of pressure differences



Substituting Equations (9) and (10) in Equation (8)

$$l_1 = \frac{R}{2R + 1}\,l \qquad (11)$$

It follows, with reference to Fig. 2(b) that

$$h'_o = \frac{2R}{2R + 1}\,h_o \qquad (12)$$

where

h'_o = pressure difference across bottom of outer pane with window above neutral zone, in. of water

Also

$$h'_i = h - h_o \frac{1}{2R+1} \qquad (13)$$

where

h'_i = pressure difference across center of inner pane with window above neutral zone, in. of water

h = inside-outside air pressure difference across center of window, in. of water

Vapor transfer comparison—To assess the relative rates of vapor flow by diffusion and air flow into and out of the air space of the test window, outside conditions of 0 F and 85% relative humidity and inside conditions of 70 F and 25% relative humidity have been chosen. The vapor pressure in the space has been taken to correspond to saturation at the temperature of the inside surface of the outer pane. This temperature was calculated assuming an outside surface conductance of 4.0 (for a 10 mph wind) and a U value of 0.51. For these conditions R in Equation (3) equals 3.42.

Based on the measured diffusion coefficient, the vapor transmission by diffusion through the cracks around the inner pane is 0.23 grains per hr. The vapor transmission by diffusion through two ¼-in. vent holes, assuming one at the top and bottom, is 0.0019 grains per hr. Calculated vapor transmission rates by air flow for different conditions of inside-outside total pressure differences are given in Table III. These values are based on the measured air flow characteristics and the simple relationships developed in the preceding section.

It can be seen that the relative importance of vapor transmission by diffusion and air flow through cracks around the inner glazing depends to a considerable extent on the inside-outside total pressure difference. With the neutral zone at the center of the window, the vapor transfer by air flow is about twice that by diffusion; with the neutral zone at the bottom edge the ratio is about 10 for the conditions chosen. At higher total pressure differences, vapor transfer by diffusion becomes relatively insignificant.

The vapor transfer through the ¼-in. vent holes by diffusion is unimportant compared to that by air flow, regardless of the total pressure difference, so long as the vents are distributed so that chimney action can occur between the air space and outside.

The amount of venting of the air space to outside required to maintain vapor outflow equal to inflow, and thus to avoid condensation between the panes, can be determined from Table III for the conditions chosen. With the neutral zone at the center of the window, 2 or 3 vent holes at top and bottom are required depending on the contribution of diffusion to vapor transfer into the space. With the neutral zone at the bottom edge 10 vent holes at top and bottom are required.

At pressure differences of 0.004 and 0.014 respectively, some 20 and 60 ¼-in. vents are required. These pressure differences were chosen to represent those that might occur across leeward windows with winds of 5 and 10 mph. If vapor transfer from the air space to outside by diffusion is the only available mechanism, the number of vents required is many times the values just given.

Air flow into and out of the window space as a result of "breathing" action due to temperature changes is another possible means of vapor transfer. It can be shown for the test window that the average air flow into or out of the space due to a daily outside temperature cycle from −20 F to +20 F is about 0.001 cu ft per hr. This compares with a calculated air flow through the cracks around the inner glazing as a result of chimney action, with the window at the neutral zone, of 0.42 cu ft per hr.

Thus, for the test window, the amount of vapor transfer as a result of "breathing" action is insignificant relative to that as a result of chimney action. The importance of "breathing" action in moving water vapor into or out of the air space of a double window will depend on its air leakage characteristics and the volume of air between the panes. Unless the air space is unusually thick, "breathing" action will have only a secondary effect on all but extremely tight windows.

Cold room tests — In predicting for the test window the amount of venting of the air space to outside required to overcome condensation under specified conditions, a number of simplifications were introduced. For example, in determining the stack action between the air space and outside it was assumed that the air space temperature could be taken as equal to the mean of the temperatures of the two panes. Both horizontal and vertical temperature gradients in the space were disregarded.

Cold room tests were undertaken to determine by direct observation the amount of venting required under specific conditions. For this study, a test window was installed in a wall panel which formed part of a partition between a warm and cold room. The temperature of the cold room could be controlled over a wide range within ±0.25 F, with floor to ceiling temperature gradients not exceeding 1 F. The relative humidity in the cold room was about 50%. Air movement over the window was downward at about 6 mph. Close control of temperature and humidity was also provided in the warm room.

With a warm room temperature of 70 F at the window level, observations of condensation between the panes were made at a number of cold room temperatures and warm room humidities, with various degrees of venting of the air space to the cold room. Results obtained with a cold room temperature of 0 F and a warm room humidity of 38% are given in Table IV. None of the conditions covered by Table IV led to more than light deposits of frost. For the two venting conditions judged adequate, the frost was limited to small patches in the two upper corners. The vents were located near the center of the top and bottom edges. It is thought that if they had been more uniformly dispersed the window would have remained entirely clear. For the two venting arrangements judged to be borderline, there were two or three small additional patches of light frost near the bottom of the window. With the openings at top or bottom only, there were two light bands of frost from top to bottom on either side of center.

Measurements were made to determine the air flow characteristics of the cracks around the inner pane of the window unit used for the cold room study and it was found to be somewhat tighter than that used for the vapor diffusion measurements.

Measurements of pressure difference between cold and warm rooms were made to determine the total pressure distribution over the test window. Results of these measurements with the cold room at 0 F, given in Fig. 7, indicate that the neutral zone was between the center and bottom edge of the window.

No measurements of the vapor diffusion characteristics of the cracks around the inner glazing used in the cold room tests were made. Predictions of the amount of venting required to prevent condensation between the panes must therefore be based on air flow alone. Such calculations show that with the neutral zone at the center of the window, 1½ vent holes top and bottom are required, while 4 vent holes top and bottom are needed with the neutral zone at the bottom edge of the window. The vapor pressure in the air space was assumed to correspond to saturation at the minimum temperature measured on the inside surface of the outer pane.

The comparison of vapor transfer by diffusion and air flow for the inner glazing used in the diffusion measurements has shown that diffusion contributes a significant proportion of the vapor flow at these small pressure differences. Calculations for this unit under the cold room conditions show that the flow by diffusion is from one-third to one-half that by air flow, with the window at the neutral zone. Applying this proportion to the window unit tested in the cold room, the number of vent holes required becomes 2 and 4½, with the neutral zone at the center and bottom edge respectively.

There is fair agreement between the cold room observations and the estimated amount of venting of the air space required to overcome condensation between the panes. The cold room tests were not ideal for such a comparison because of the quite small total pressure differences provided, when accurate measurements of pressure and flow become difficult. Furthermore, small changes in the location of the neutral zone will lead to relatively large changes in the venting requirements.

It is clear that cold room studies of condensation between the panes of double windows are of little value unless the total pressure differences across the window are measured. Even then, the observations are of use only in confirming some other approach for determining condensation performance, unless the total pressure differences correspond to those for which the window is being considered.

Venting – It has been shown that venting of the air space to outside through chimney action is an effective means of controlling condensation between the panes of double windows. It will be recognized, however, that excessive venting will lower the mean temperature of the air space and increase over-all heat transmission. Taking the air space as equivalent to two surface conductances, its mean temperature can be estimated by equating the heat flow through the inner pane to the heat flow through the outer pane plus that carried away by the ventilating air. The apparent over-all heat transmission coefficient with venting can then be expressed as

$$U' = U \, \frac{T_i - T_a'}{T_i - T_a} \qquad (14)$$

where

U' = apparent over-all heat transmission coefficient with venting

U = over-all heat transmission coefficient without venting

T_i = inside air temperature

T_a' = mean air space temperature with venting

T_a = mean air space temperature without venting.

On this basis the percentage increase in the over-all heat transmission coefficient has been calculated with venting as required at 0 F and 85% relative humidity outside, and 70 F and 25% relative humidity inside.

For the test window used in the diffusion measurements, the increase in U value is 1% with a total pressure difference of 0.004 in. of water, and 2.5% with a total pressure difference of 0.014 in. of water. The increase in U value for the window used in the cold room studies is less than 1% at a total pressure difference of 0.014 in. of water. Both these specimens are, however, relatively tight. At a pressure difference of 0.301 in. of water (equivalent to the velocity pressure of wind at 25 mph) the air leakage rate of the latter is 0.04 to 0.05 cu ft per min per ft of crack.

Leakage through the other specimen was not measured at this pressure difference. It can be estimated, however, assuming that the flow can be expressed as

TABLE III

CALCULATED VAPOR TRANSFER BY AIR FLOW FOR WINDOW USED IN DIFFUSION TEST

Total Pressure Difference (in. of water)	Vapor transfer through cracks around inner pane (grains per hr)	Vapor transfer through one ¼-in. vent top and bottom (grains per hr)
neutral zone at center $h = 0$	0.515	0.29
neutral zone at bottom edge $h = 0.002$	2.34	0.26
$h = 0.004$	5.1	0.26
$h = 0.014$	15.7	0.26

TABLE IV

STUDIES OF WINDOW VENTING IN COLD ROOM

Room Conditions	Duration of test (hr)	Arrangement of ¼-in. vent holes	Observations
Cold room at 0 F	26	3 top and bottom	borderline
Warm room at 70 F and 38% relative humidity			
	74	4 top and bottom	borderline
	92	5 top and bottom	adequate
	124	6 top and bottom	adequate
	90	23 bottom	inadequate
	24	23 top	inadequate

$$Q_i = c\ (h)^n \qquad (15)$$

where

c = flow coefficient

n = exponent of flow, between ½ and 1

The exponent n is about 0.55 for the test units. The flow through the specimen used in the diffusion tests is then 0.07 to 0.08 cu ft per min per ft of crack at a pressure difference of 0.301 in. of water.

These leakage rates are probably representative of the leakage through cracks around the inner pane of many residential and commercial windows of the casement, awning, or hopper types, having inner and outer glazing in the same sash. There are other window types having a much lower resistance to flow from inside to the air space. For example, the Specification of the Aluminum Window Manufacturers Association permits a maximum air leakage rate of 0.5 cu ft per min per ft of crack at a pressure difference of 0.301 in. of water for a double double-hung window.[3]

An approximation of the venting requirements and the resulting increase in U value for such a window, for the conditions just referred to, can be made by extrapolating flow rates on the basis of Equation (15). If it is assumed that all the resistance to flow is from inside to the air space, the increase in U value is about 9% at a pressure difference of 0.004 in. of water and about 16% at a pressure difference of 0.014 in. of water. If it is assumed that the resistance to flow is normally evenly divided between inner and outer sash the increases in U value are 13% and 21% respectively with the required venting. The advantage of a tight inner sash is apparent.

In applying principles of venting to the design of a window, ambient conditions should be carefully selected. It is usually not necessary to design for the most extreme condition, since some condensation can generally be tolerated and time is required for its accumulation. Since information on the actual total pressure differences across building walls is quite limited, application of venting principles should be combined with direction observations of field performance.

CONCLUSIONS

With outside temperatures lower than inside, condensation will ultimately occur on the inside surface of the outer pane of double windows if the gain in water vapor to the window space is greater than the loss.

Vapor pressure differences available for diffusion of water vapor into the air space are several times greater than those available for diffusion from the air space to outside, the ratio of the pressure differences increasing with decreasing outside temperature.

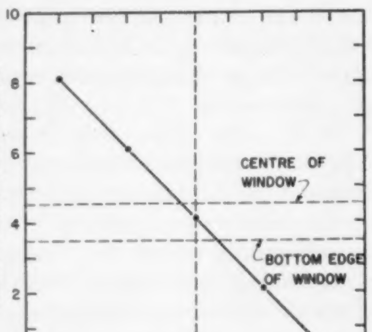


Fig. 7 With the cold room at 0 F

Vapor transfer as a result of air flow will depend on the total pressure difference across the window and the relative resistance and distribution of cracks around inner and outer panes as well as psychrometric conditions.

For windows having air leakage rates equal to or greater than the test windows (0.05 cu ft per min per ft of crack at 0.301 in. of water), the vapor transfer by air flow will probably be several times that by diffusion even with the window at the neutral zone, the ratio increasing as the height above the neutral zone increases.

With the air flow resistance of the inside pane sufficiently higher than that of the outside pane, and with openings around the outside pane located at the top and bottom of the air space, the air space can interchange air with the outside by "chimney action" even with inside pressures greater than outside. Such venting of the space can be used effectively to prevent condensation between the panes if the amount of outside air interchange is large in relation to the air flow to the space from inside. The ratio of outside to inside air required to prevent condensation depends only on inside and outside psychrometric conditions.

The number of vents required to prevent condensation for given design conditions can be predicted from simple pressure relationships, providing the air flow characteristics of the vents and the cracks around the inner pane are known.

Excessive air interchange between the air space and outside can result in significant increases in the over-all heat transmission coefficient. It may therefore be impractical to apply venting to windows having high leakage rates.

There is a lack of detailed information on actual conditions in buildings that affect condensation on the inside surface of the outer pane. Specifically, further information on total air pressure differences between inside and outside, and inside relative humidities, is desirable.

Unless conditions of test, including total air pressure differences, are similar to those expected in the field the results of observations of condensation on the outer pane in cold room studies are not directly applicable and are of doubtful value except for research purposes.

ACKNOWLEDGMENT

The authors are indebted to Dr. D. G. Stephenson for assistance in the measurement of air pressure differences with the displacement-type micromanometer, to K. R. Solvason for assistance in the cold room studies and to M. O. Pelletier, laboratory assistant, who assembled the equipment and took most of the records. The advice of Dr. N. B. Hutcheon, Assistant Director of the Div. of Building Research gratefully is acknowledged.

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Ground is broken for United Engineering Center

As Herbert Hoover turned the token spadeful of earth that will lead to the construction of the multi-society United Engineering Center in New York. He is here flanked by Jerry Fujimoto, student from Hawaii representing engineers of the future and President Andrew Fletcher of United Engineering Trustees.

Despite generous buffetings and drenchings by Hurricane Gracie, ground was broken for the United Engineering Center in New York on October 1 before a distinguished audience of engineers and their guests.

Where UEC shortly will rise between 47th and 48th Streets on the west side of United Nations Plaza, several hundred individuals gathered at 11:00 a.m. in a city lot cleared of all structures but without evidence of forthcoming developments. Ground was broken by ex-President of the United States Herbert Hoover, following brief talks by President Andrew Fletcher of United Engineering Trustees, Chairman Willis F. Thompson of the Real Estate Committee, Chairman R. E. Dougherty of the Member Gifts Campaign Committee, and Chairman Mervin J. Kelly of the Industrial Building Fund Committee.

The UEC building to be readied for occupancy in 1961 will be an 18-story structure with a gross area of 263,000 sq ft and a net area of 180,000 sq ft. ASHRAE will occupy the fourth floor on the southern half of the initial structure.

As indicated previously, pages 54, 55 and 56 of the ASHRAE JOURNAL October 1959, a membership campaign to raise funds for the partial financing of the structure is well under way. At least one society has exceeded its quota already and, according to Chairman Dougherty, member gifts total 76% of that aggregate agreed-upon obligation.

Gifts from industry were reported upon encouragingly by Chairman Kelly.

ASHRAE's fund raising campaign under the direction of Chairman Merrill F. Blankin started this same day and it is anticipated that the quoted $250,000 will have been raised with reasonable promptness, perhaps by the end of October.

Ex-President Hoover and other speakers touched more or less repetitiously upon some of the broader aspects of the significance of the United Engineering Center, emphasizing the importance of engineering in the world in which we live, recognizing the essential encouragement that is required for younger engineers and paying tribute to the initial generosity of Andrew Carnegie who made possible the first Engineering Societies Building in New York, long since outgrown and outmoded for the needs of today.

Conspicuously adding to the occasion were such guests as Alfred P. Sloan, Jr., former chairman of General Motors Corporation, and the presidents of most of those societies which will occupy the premises. President Arthur J. Hess of ASHRAE was among those present as was ASHRAE-UEC Fund Raising Committee Chairman Merrill F. Blankin. World-wide representation was included among the guests and delegates to the occasion.

The new Center will include one of the largest engineering reference libraries in the world, if not the largest, will provide reference facilities upon paralleled extent, will provide prompt duplication facilities and will house the 600 headquarters staff of 19 participating societies.

Speaking at the ceremony, Mr. Hoover observed:

"This is an event of national importance.

The Engineering Societies in our country compose a membership of over 250,000 men and women covering almost every branch of the profession. They are the foundation of security in our defense and the increase of our standards of living and comfort.

The Engineering Societies are in reality a great educational institution. Within the Societies is the constant exchange of discovery, improvements and experience. Their findings are printed for all the world to see.

These Societies also constitute a gigantic post-graduate course for engineers. It continues all their professional lives. I have attended this course for more than sixty years with intellectual profit.

I have said before now that the job of the engineer is to take from the scientists their discoveries and from the inventors their findings, and to apply them for the use of the people everywhere. The engineer starts with these stimulants to his imagination. He makes a plan on paper. Then he moves to its realization in cement, in metal, in stone, and in energy. Thus he brings jobs and better homes. This is high privilege among all professions.

The purpose of this great building is to facilitate these goals. It will play a great part in American life. It will serve all mankind."

They Wanted to Know

Inquiries of the month to ASHRAE Headquarters covered many points as to technical facts, standards, practices, personnel and published references. From these, the following have been selected and condensed as being those replies of some general interest and value to ASHRAE members.

ELECTROMAGNETIC COMPRESSORS

To ASHRAE:

Enclosed is a photograph and a schematic layout of an electromagnetic compressor, as manufactured overseas. Is this unit, or a corresponding one, being manufactured in the U.S.?

H. M. D. N.

We are unable to verify that such a machine is being produced in this country. Several years ago, there was a veritable rash of patent applications on electromagnetic compressors but, due to the many practical operational problems to be met, none of the designs was successful. Starting torque requirements, plus the necessity of operating against the normal maximum discharge pressure, are high among the difficulties to be overcome. Noise, too, is a problem particularly when the piston is loaded lightly. So far as we know, none of these machines was ever put into production.

JOB GUIDANCE

To ASHRAE:

I am an American citizen, qualified as a Mechanical Engineer in Denmark, have specialized in heating and air conditioning in London and want to settle down in California. Can you assist me in contacting firms which may be interested in employing me?

C. B.

We suggest that you place a classified advertisement in the ASHRAE JOURNAL. You might also contact Indoor Comfort News, the monthly publication of the Institute of Heating and Air Conditioning Industries and the Warm Air Heating Institute of Northern California.

ONE TO ONE

To ASHRAE:

We have a turkey chilling problem and wish to determine the ice making requirements per pound of product. What is the accepted ratio?

H. L. R.

It takes one pound of ice slush to chill one pound of eviscerated poultry. The weight of ice actually required to cool a turkey carcass (specific heat 0.8) at 100 F (maximum initial temperature) to 32 F is about 0.4 lb. Remaining allowance is used to cool the water to make up the ice slush, to allow for heat gain from the air around the tank, for air used for agitation and to allow for ice discarded at the end of chilling. Lowering the surrounding air temperature and precooling the water used in making the ice slush will influence ice requirements. In the absence of any precooling, the natural temperature of the tap water will influence ice requirements. For each 10 F decrease in entering tap water temperature the ice requirements will be reduced 0.6 lb ice per gal of water used.

WHY NOT RECOMMENDED

To ASHRAE:

In the 1958 issue of the ASHRAE GUIDE it states "The practice of connecting a tempering tank with a firebox coil, the electric heater then furnishing heat only when there is deficiency from the boiler or furnace, is not recommended." I would like to know the reasons for this statement.

D. C. R.

The objection to a firebox coil is that it continues to draw heat from a combustion chamber regardless of the water temperature in the tank. The erratic and uncontrolled operation of a firebox coil is the reason for objecting to it. Residential hot water system faucet temperatures are kept at or below 140 F because iron and steel pipe and steel tanks are corroded readily above that, particularly if the water is high in oxygen content.

ELECTRICITY OR GAS

To ASHRAE:

We are interested in installing a new heating and cooling system in our store. We have talked with salesmen, promoting natural gas jobs as well as those who handle electrical systems. Of course, each salesman claims his to be the best. Can you advise me upon the pros and cons?

R. J. D.

There are single systems which both heat and cool and may be operated by electricity or by gas. There are separate heating and cooling systems operated by either fuel. Thus, your problem is essentially one of economics since it may be assumed that adequate performance is available in either type. You should compare first the cost of installation and second the operating cost. To a degree, both of these factors are related to local conditions. We suggest that you contact your electric utility and gas utility in your community where a staff engineer should be available to give you the economic factors. You might also write the Edison Electric Institute for booklets or reports upon comparative cost and direct a similar inquiry to the American Gas Association.

HARTFORD LOOP

To ASHRAE:

We have a technical disagreement in our office concerning the Hartford loop. One viewpoint holds it to be a means of equalizing pressures in a steam vapor system. The other is that the loop has the primary function to allow sufficient water to remain in the boiler where it warns by hammering.

G. S.

The horizontal portion of the Hartford connection is made as short as possible to prevent water hammer; the connection is not used for the purpose of obtaining a warning noise. If there is a leak in the return line to the boiler, with a Hartford connection the boiler will be safe until enough water evaporates to bring the water surface below the level which exposes the boiler surface not in contact with the water to the hot gases. It serves also to prevent the pressure in the boiler from pushing water out of the boiler into the return.

NO BRASS WITH AMMONIA

To ASHRAE:

Claims are made that it is dangerous to use brass fittings for ammonia refrigeration systems, why? I work in a dairy which uses such a system.

E. F. C.

Use of brass fittings in ammonia systems is not recommended. If such a system can be completely anhydrous there will be no ill effects. However, if any moisture whatsoever is introduced, corrosion will take place on the brass fittings. For such systems special fittings have been developed and should be used. Though no corrosion effect is noticeable on your system as yet, it is inevitable that this will take place and eventual trouble will cause replacement of such fittings.

HOW SMALL

To ASHRAE:

It is rumored here in Australia that there is an amazingly small, highly efficient new type of air conditioning compressor being perfected in your country. We have been told that the unit is comparable in size to an ordinary tea cup.

W. S.

While it is true that a number of new designs released recently and about to be released are constantly reducing size and weight as well as cost, we know of no compressor of the size of an ordinary tea cup. Engineers tell us that within the next few years, the size of a teapot might be approximated in a practical mass produced unit.

ASHRAE STANDARDS COMMITTEE

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42P — **Standard for Comfort Air Conditioning**
Chairman: M. K. Fahnestock
Liaison: S. P. Soling

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Members: S. F. Duncan, H. S. Dutcher, E. F. Gritton, Leo Hungerford, C. B. Rowe, W. T. Smith, J. R. Watt

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Members: W. C. L. Hemeon, R. F. Logsdon, P. E. McNall

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Members: G. C. Breidert, F. N. Calhoon, R. B. Engdahl, W. V. Hukill, J. P. Johnson, K. E. Robinson, F. B. Rowley, L. J. Shumaker

Meetings ahead

November 1-2 – Air-Conditioning and Refrigeration Wholesalers, Annual Meeting, Atlantic City, N. J.

November 2-5–11th Exposition of the Air-Conditioning and Refrigeration Industry, Atlantic City, N. J.

November 9-13 – National Electrical Manufacturers Association, Annual Meeting, Atlantic City, N. J.

November 17-19 – Building Research Institute Semiannual Conference, Washington, D. C.

November 9-13 – Institute of Boiler and Radiator Manufacturers, Semiannual Meeting, Absecon, N. J.

December 3-4 – National Warm Air Heating and Air Conditioning Association, Annual Convention, St. Louis, Mo.

December 26-31 – American Association for the Advancement of Science, Annual Meeting, Chicago, Ill.

February 1-4 – American Society of Heating, Refrigerating and Air-Conditioning Engineers, Semiannual Meeting, Dallas, Texas.

February 1-4–2nd Southwest Heating and Air-Conditioning Exposition, Dallas, Texas.

March 6-10 – National Association of Frozen Food Packers, 19th Annual Meeting, Chicago, Ill.

March 21-23–First National Electric House Heating Exposition, National Electrical Manufacturers Association, Electric House Heating Equipment Sect, Chicago, Ill.

April 27-30 – 3rd Western Air-Conditioning, Heating, Ventilating and Refrigeration Exhibit and Conference, Los Angeles, Calif.

May 1-4 – Air-Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

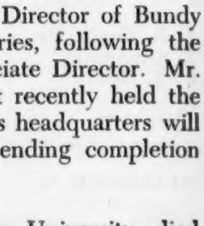
May 19-21 – Refrigeration Research Foundation, Annual Meeting, Denver, Col.

June 13-15–American Society of Heating, Refrigerating and Air-Conditioning Engineers, 67th Annual Meeting, Vancouver, B. C.

People

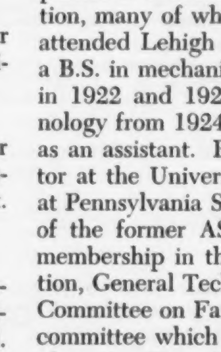
Arthur F. Bernthal succeeds to the position of Managing Director of Bundy Tubing Company's Research and Development Laboratories, following the retirements of the former Director and **G. S. Wiley,** Associate Director. Mr. Bernthal, with the company for more than 25 years, most recently held the position of advertising and sales development manager. His headquarters will be at the Research Laboratory at Birmingham, Mich., pending completion of the company's new facilities at its Detroit plant.

John E. Nicholas, Professor Emeritus of Pennsylvania State University, died recently, ending a 30-year career of teaching and research. A prolific writer, Prof. Nicholas was the author of more than 75 books, articles and reports on phases of mechanical engineering and dairy farm electrification and refrigeration, many of which appeared in Refrigerating Engineering. Born in 1893, he attended Lehigh University from 1911 to 1915, receiving a B.S. in mechanical engineering, the University of Illinois in 1922 and 1923, and Massachusetts Institute of Technology from 1924 to 1926, receiving his M.S. and working as an assistant. From 1926 until 1929 he was an instructor at the University of Minnesota, when he left to teach at Pennsylvania State College (later University). A Fellow of the former ASRE, Prof. Nicholas, in his 30 years of membership in the Society, was a member of the Education, General Technical and Standards Committees; Chairman of the Technical Committee on Farm Refrigeration, Food Freezer Standards Committee and the committee which prepared the code on "Methods of Rating and Testing Complete Can-Type Milk Coolers"; and ASRE representative on the Farm Applications of Refrigeration Committee in 1946.



William H. Krapohl, owner since 1951 of William H. Krapohl and Associates, Boston, Mass., died on August 12th at the age of 60. A member of the former ASHVE since 1941, Mr. Krapohl had been engaged in the fields of heating and ventilating since his graduation from Northeastern University in 1919.

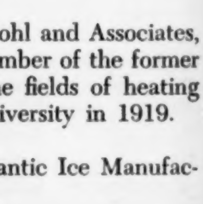
Daniel J. Montague was Assistant General Manager of Atlantic Ice Manufacturing Company, Coatesville, Pa., before his recent death.



Stanley R. Curtis, an engineering graduate of Michigan State University, has been appointed Project Engineer for Tranter Manufacturing, Inc. Vice President of the Western Michigan Chapter, ASHRAE, Mr. Curtis was formerly Senior Project Engineer and Group Leader in the gas heating appliance department of Motor Wheel Corporation. He also served in the insulation department and as a residential and industrial works estimator.

Olan C. Reese died in September, after more than 20 years of work in the refrigeration industry. He was a partner in Charles A. Myers and Company.

Harold P. Mueller, Sr., President and General Manager of Mueller Climatrol, Div of Worthington Corporation, was elected to the Board of Directors of the Milwaukee Gas Light Company on September 15th. Mr. Mueller, a grandson of the founder of L. J. Mueller Furnace Company, joined Mueller Climatrol in 1919 as a salesman, was promoted to Sales Manager and named Vice President in Charge of Sales in 1929. He became President and Treasurer in 1931. Upon acquisition of the company by Worthington in 1954, Mr. Mueller became President and General Manager of the Division and a director of Worthington. President of the National Warm Air Heating and Air Conditioning Association in 1944 and 1945, he is an active member of the Gas Appliance Manufacturer's Association and ASHRAE.



A. E. Diehl, Project Engineer for York Div of Borg-Warner Corporation, has been named Assistant Manager of the Division's Hartley Plant in York, Pa., and will assume immediate supervision over special projects carried on there. Mr. Diehl was employed by the company following his graduation from Gettysburg College in 1933 with a degree in mechanical engineering. He has served as test operator, test engineer, analyst and application engineer, development engineer, sales engineer, engineering manager in charge of self-contained and residential air conditioning development and chief engineer of the Commercial Div. Membership Chairman of the local chapter of ASHRAE, Mr. Diehl is engaged in writing a chapter on air cooling and heating coils for the new ASHRAE Data Book.

Herman C. Koenig had served with Electrical Testing Laboratories, Inc., from 1915 until his death in August, and as an instructor at Cooper Union Night School of Engineering between 1916 and 1951. Born in 1893, Mr. Koenig was a Fellow of the American Institute of Electrical Engineers and a Member of the former ASRE since 1953.

Frank D. Klein, a member of ASHRAE, the American Society of Mechanical Engineers and the American Society for Testing Materials and the author of numerous textbooks and technical papers in the refrigeration field, has been appointed General Manager of Dunham-Bush's Michigan City, Ind., plant. Mr. Klein comes to Michigan City from the company's main office in West Hartford, Conn., where he was Product Manager–Air Conditioning.

Leon T. Mart, founder of Marley Company and President for the 37 years of its operation, died in June of this year at the age of 68, only a few months after being elected Chairman of the Board of Directors. A graduate of Lehigh University, he was the author of several technical papers and held more than 25 American and foreign patents. Elected President of the Cooling Tower Institute in 1953, he had served on the Council of the former ASHAE and was recently named a Fellow of ASME.

Lawrence A. Philipp retired on October 1st from his position as Vice President in charge of appliance engineering and research with Kelvinator Div of American Motors Corporation. During his 32 years with the company, Dr. Philipp was issued 186 patents. His contributions to the advancement of product design and performance in the major appliance industry included the cold-clear-to-the-floor design introduced in 1948.

Joseph M. Aikman, owner of Golden Triangle Mechanical Service, passed away on August 29th, at the age of 67. A member of the Florida West Coast Chapter of ASHRAE, Mr. Aikman had been involved in heating and ventilating since 1910. Major projects on which he worked were design of the boiler plant for the Pentagon, heating and air conditioning for the War Department Building and air conditioning for the White House.

Morris Backer, a mechanical engineering graduate of Texas A & M, has been promoted to the position of Associate in the firm of H. E. Bovay, Jr., Consulting Engineers. A member of the Texas Society of Professional Engineers, Mr. Backer has been with the company since 1952, engaged in mechanical design of air conditioning, heating, ventilating and plumbing systems for commercial and industrial installations.

Robert M. Krohmer, prior to his death in August, was Project Engineer with Armstrong Furnace Company. Educated at Ohio State University, he had been employed by Chrysler Air Temp Div from the time of his graduation until he joined Armstrong.

Others are saying—

that . . . . . . development of a filtered air ventilator which can be applied to Airslide bulk rail cars, bulk flour trailers and barges is cited as being the first successful attempt to solve the problems occurring during transportation of bulk flour. An intake ventilator is installed at the top of the break-end of the car, equipped with a filter and protected by a snow baffle. Connected with it is a fan which pulls 500 cfm of filtered air through the car at all times while it is in transit. On the opposite end of the car is mounted a 10-in. fan for expelling the air. Inside the car at each of these vents are baffles which direct the intake and exhaust of the air toward the walls and roof of the car so that the full area above the flour level is changed constantly. The fan operates from the time the hatches are sealed at the mill until the car arrives at the bakery and from the time it leaves the bakery until it returns to the mill, running continuously to keep the entire car dry. Keeping the car quite dry during transit, which this ventilation system accomplishes, is, however, not the sole solution to the problem, as the effectiveness of the ventilator for the end results is dependent upon a satisfactory method of loading that will keep the entire car surface dry during this process. *Baker's Digest, August 1959, p 63.*

that . . . . . . refrigerated warehouses with insulation fixed to the inside of the exterior walls may have considerable losses where this insulation is intersected by floors and dividing walls. Measures successfully used to reduce such heat loss were: reduction of thermal conductivity of the floors (by changing the composition of the flooring materials), insulated joints in the exterior walls, insulation fixed to the outside of the exterior wall, and application of local heating to neutralize cold flow causing surface condensation in adjoining heated rooms. *Kulde, Danish Journal of Refrigeration, August 1959, p 68.*

What ASHRAE Chapters are doing

Formation and development of the Society since the merger and role of the individual chapters in the national organization keynoted speeches at opening meetings of the 1959-1960 season, members of two groups being addressed by ASHRAE President Arthur J. Hess. Trends in air conditioning and heating headed the wide range of technical topics, with panel discussions prominent.

LOUISVILLE . . . Opening their first season as a merged group, members of this chapter heard James W. May, ASHRAE Region VII Director, speak on the "Function and Operation of the National Society." Mr. May outlined how the society is set up to serve the individual member, the appointment of committees, operation of the research laboratory, cooperative research programs at colleges and universities, allotment of the budget and the headquarters staff and its function.

Presented for discussion at this meeting was a petition brought forth during the summer to have this chapter transferred from Region VII to Region V. No vote was taken on the petition, purpose of discussion being to inform the members that it had been made and to permit exchange of views. Chapter membership will be polled by the national headquarters to determine whether the petition is to be presented to the Board of Regional Directors for action.

MIDDLE TENNESSEE . . . Presentation of the Charter to Chapter President I. C. Thomasson by ASHRAE President Arthur J. Hess highlighted the September 15th meeting, at which Mr. Hess delivered an address on "Role of the Chapters in ASHRAE Operation".

NEW ORLEANS . . . Members of this chapter were guests at the Southern Bell Telephone and Telegraph Company on a tour of the main office communications facilities. Of particular interest to the group was the system providing filtered, conditioned air with a high degree of humidity control to the areas housing delicately adjusted automatic equipment. Because of the basic layout of the building, the air conditioning is handled by seven separately located zoned stations rather than one central plant.

Following the tour, members heard Samuel Weinstein, Division Plant Supervisor of the company, explain the mechanical functions of the several types of equipment used in handling both long distance and local service. Waring Green then gave a live demonstration of microwave transmission and the direct long distance dialing system.

CENTRAL ARIZONA . . . An address on "The Economics of Complete Air Conditioning for Schools," delivered by Zeph Marsh, School Market Sales Manager of Minneapolis-Honeywell Regulator Company, began the season for this group at its September meeting. Directly involved in the problems of school air conditioning in his work with the School Facilities Council, a national organization of school officials, Mr. Marsh was able to gather much information about school economics. Major points cited in the talk were that the air conditioning industry should have, as one objective, the betterment of school facilities and, in line with this, the industry has a responsibility to make comfort conditions prevalent in the minds of responsible school officials. Mr. Marsh accentuated his talk with illustrative slides and models showing several designs for air conditioned schools and indicating how design of school plants can be simplified by the use of air conditioning.

RICHMOND . . . "Engineers must do a better selling job," National President Arthur J. Hess informed members in a talk presented at the September meeting of this chapter. In his address, "Titles Don't Make Engineers", Mr. Hess explained his statement by further declaring that owners must be sold on having complete systems installed, architects on furnishing the engineer needed space for a complete system and young engineers on this profession.

Speaking at the September meeting of the Richmond Chapter, ASHRAE President Arthur J. Hess urged greater owner-architect-engineer cooperation. Shown here with Mr. Hess are George J. Wachter, Richmond Chapter President, and Jack Spencer, Norfolk Chapter President

EVANSVILLE . . . Region V Director T. G. Crider told those attending the September meeting about the formation and development of ASHRAE since January 29, 1959, enumerating problems existent in the reorganization. Reviewing the purposes and functions of regional operation, Mr. Crider explained it in relation to the chapter and national organizations.

WISCONSIN . . . Program Committee Chairman John Illingworth announced plans for a program schedule to include two speakers at each meeting, the preliminary speaker to be a local member giving a short (20 min) talk, then gave the first of these talks himself, on "Contractor Aspects of Our Business". Cov-

ering the various phases each contractor must go through when bidding on jobs, he summarized some of the difficulties which may hinder the contractor in completing his contract.

Guest speaker of the evening was Fred Wood, District Manager of the Chicago office of York Corporation, who spoke on "Economics of the Air Source Heat Pump". Cautioning that use of the heat pump is limited to applications where heating load is lower than or equal to cooling load, Mr. Wood stated that economically it may reflect a higher initial investment, however, lower operating costs warrant consideration. In summarizing his talk he reviewed advantages of the heat pump such as year-round operation, no heating unit or fuel, less space required and fewer operating hazards.

JOHNSTOWN . . . Various systems and their components for hydronic heating and cooling in contemporary homes were discussed by speaker of the evening Frank Crawford of White-Rodgers Company, explaining the operation, advantages and benefits of zone systems and valves. Following the talk, given at the September 8th meeting, was a question and answer period.

MONTREAL . . . "Latest Trends in Residential Air Conditioning", a talk delivered by Frank Thomson, Sales Manager of Lennox Industries Ltd., opened the season for this group at its September meeting.

SOUTHERN CALIFORNIA . . . Discussed in panel fashion by three speakers at this group's September meeting was "Condensing Methods". James Thompson, speaking on cooling towers, discussed basic definitions, types of fill, type of tower, wind effects, design of tower basin, noise levels, water treatment requirements and economic analysis.

Presented in detail by Harold Halls was the subject of evaporative condensers. His talk included information on heat rejection rates for various refrigerants, refrigerant liquid rates, condensing temperature limits, sub-cooling coils, internal surface areas, bleed-off rate and closed circuit evaporative coolers.

Final portion of the session was devoted to air-cooled condensers, as discussed by Red Butler. His talk related to limitations of air-cooled units, cost comparisons, ambient temperature conditions, method of ratings, fan selection and head pressure control.

FORT WORTH . . . Using color slides to illustrate his subject, James Mays of Sporlan Valve Company addressed the September 16th meeting of this chapter on "Theory, Application and Installation of Thermal Expansion Valves".

NEW YORK . . . Design and construction of a structure the size of the Union Carbide Building presented many interesting problems, not the least of which was the fact that the building was erected over the New York Central railroad tracks, according to Arnold L. Windman of Syska and Hennessy, Inc., Design Supervisor of this project. In his discussion, "Air Conditioning the Union Carbide Building", Mr. Windman touched on such widely diversified aspects of the project as site conditions, methods used to meet office partition flexibility requirements and research and development of designs for the building, giving major emphasis to the refrigeration plant and air conditioning systems.

COLUMBUS . . . Instituted at the first meeting of the 1959-1960 season, held on September 21st, was a new policy of having two speakers for the evening, one to deliver a short (10 min) "Coffee Talk" preceding the longer address of the main speaker.

Thomas B. Simon, Product Evaluation Manager of Westinghouse Electric Corporation, gave the Coffee Talk on "What Makes the New Frost-Free-Type Refrigerator Tick".

"Comfort Conditioning for Space Travel" was the subject of a lecture given by Dr. Fred A. Hitchcock of the Department of Aviation Physiology of Ohio State University, main speaker of the evening, basis of whose talk was the idea that when man begins his travels into space he will require means to maintain his comfort, his performance efficiency being greatly dependent on his environment. Following a brief rundown on the technical phases and the complexity of the recent Russian moon rocket, Dr. Hitchcock proceeded to discuss the various physiological aspects of orbiting a man around the earth. While outlining the weight penalty involved in supplying oxygen to humans on such a flight, he stated that one cu ft of the most active algae in ten cu ft of culture water would furnish enough oxygen and absorb enough carbon dioxide to supply the needs of 100 men.

BALTIMORE . . . Inspection of a natural gas engine system, used to air condition the meeting place of this group, complemented the speaker's topic, "Natural Gas Engines as Applied to Air Conditioning". Norbert Hall of the Ready-Power Company delivered the talk.

NORTHERN CONNECTICUT . . . Nine forthcoming meetings were announced at the first session of this newly combined and geographically redistributed chapter, to include plant tours to Combustion Engineering, Inc., at Windsor Locks, and Electric Boat Company at Groton.

Acting as moderator for a panel of four speakers, each covering a phase of instrumentation concerning the air conditioning field, was Fritz Honerkamp. Thermometry and psychrometry were discussed by Raymond Barlow; pitot tubes, venturi meters and orifices by Elliot Godes; anemometry by R. I. Miller; and noise measurement by Peter Baade. Each speaker demonstrated several typical instruments, stressing the purpose, stability, accuracy and limitations of each. Opinion of the panel was that three major areas of error common to most of the instruments must be considered by the user: inherent mechanical inaccuracies, poor calibration and application errors.

ST. LOUIS . . . Quoting excerpts from publications taking opposite sides on this controversy, Gerhardt Kramer of Kramer and Harms, Architects, addressed this chapter on "The Architect Versus the Engineer as the Prime Professional".

Candidates for ASHRAE Membership

Following is a list of 122 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before November 30, 1959 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note:
* Advancement † Reinstatement

REGION I

Connecticut

CRAWLEY, P. W., JR., Sales Engr., Dunham-Bush, Inc., West Hartford.
DUNCAN, E. L., Sales Repr., Trane Co., West Hartford.
KOETTER, F. W., Sales Engr., Minneapolis Honeywell Regulator Co., Hartford.
LARSON, WILLIAM, JR., Appl. & R. & D. Engr., G. & O. Mfg. Co., New Haven.
REDDING, T. G., Appl. Engr., Dunham-Bush, Inc., West Hartford.

Massachusetts

SABOURIN, A. L., Plt. Engr., M & C Nuclear, Inc., Attleboro.
VENO, R. O., Mech. Engr., Alonzo B. Reed, Inc., Boston.
WYLLIE, D. Y., Utility Engr., M & C Nuclear, Inc., Attleboro.

New Jersey

VERSCHOOR, J. D., Market Specialist, Johns Manville, Research Center, Manville.
WILDER, C. M., Mgr. of Mfg. & Engrg., Clark Door Co., Newark.

New York

BARR, R. A.,* Sales Engr., Carrier Corporation, E. Syracuse.
EASTMAN, R. E., Jr. Proj. Engr., Brown-Lipe-Chapin, Div of General Motors Corp., Syracuse.
FERRARA, A. T., JR., Design-Draftsman, Voorhees, Walker, Smith, Smith & Haines, New York.
GLAZER, B. D., Partner, Bermac Refrigeration Service Co., Flushing.
GOLDEN, J. J., Sales Engr., Alumiseal Corp., New York.
PADOVANO, R. C., Liaison Engr., Mfr. Repr., New York.

REGION II

Canada

McCOPPEN, D. A., Mfr. Repr., Burlington, Ont.
McELNAY, J. C., Chief Engr., Greater Niagara General Hospital, Niagara Falls, Ont.
McDONALD, W. M., Br. Mgr., H. H. Angus & Assocs. Ltd., Hamilton, Ont.
MOULD, W. A.,* Secy-Treas. & Gen. Mgr., English & Mould, Ltd., Rexdale, Ont.

REGION III

District of Columbia

FAISON, T. K., JR.,* Engr., National Bureau of Standards, Washington.

Maryland

JACOBS, J. H., Mech. Designer, Henry Adams, Inc., Baltimore.

Pennsylvania

BAREIS, L. E., Htg. Repr., American Radiator & Standard Sanitary Corp., Pittsburgh.
COURTNEY, C. C., Sales Engr., Johnson Service Co., Harrisburg.
CREATI, A. J., Partner, Creati, Goren & Giannini, Inc., Abington.
GARMAN, R. B., Cons. Engr., Ebensburg.
WASHOFSKY, A. F., Proj. Engr., John J. Nesbitt, Inc., Philadelphia.

Virginia

BOYD, F. J., Control Engr., Robertshaw-Fulton Controls Co., Richmond.

REGION IV

Florida

DRIVER, J. A., Repr., Trane Co., Orlando.
KERLEY, J. J., Engr., Weeks Engineering Co., Miami.

Georgia

CHANG, V. G., Engr., Lennox Industries Inc., Decatur.
GRAY, R. S.,* Vice Pres., Refrigeration Appliances, Inc., Atlanta.
SMALLWOOD, J. W., Assoc., Felix J. Commagere, Atlanta.
SMITH, P. W., JR., Mgr., Sam P. Wallace & Co., Atlanta.

North Carolina

KESLER, H. B., JR.,† Pres., Air Engineering Co., Charlotte.
SMITH, I. J., JR.,* Factory Planning Engr., Western Electric Co., Winston-Salem.

REGION V

Illinois

BOWHAY, E. J., Technologist, Texaco Inc., Chicago.
MONTAMBO, S. T., Engr. & Estimator, Western Supply & Furnace, Maywood.

Indiana

DOYLE, D. S., Engr., Marsh Foodliners, Inc., Yorktown.
FELLWOCK, L. W., Engr., Whirlpool Corp., Evansville.
WILM, G. E.,† Test Engr., Whirlpool Corp., Evansville.

Ohio

DOENCH, F. H., JR.,* Partner, Helmig-Lienesch & Assoc., Dayton.
FLOWERS, W. J., Vice Pres., Breeding Insulation Co., Cincinnati.
HOULISTON, G. B., JR., Sales Engr., G. B. Houliston Co., Cincinnati.
NAUERT, J. P., Sales Engr., Johnson Service Co., Cincinnati.
STUTZMAN, H. A.,† Mgr. of Quality Control, Copeland Refrigeration Corp., Sidney.

REGION VI

Illinois

MARCH, H. S., Engr., Pyle-National Co., Chicago.
RATHBUN, L. J., Mfr. Repr., Peoria.

Michigan

HYDE, J. F., Engr., J. B. Olivieri & Assocs., Saginaw.
PRZYBYSZ, N. C., Repr., Hopson Bennett Co., Grand Rapids.
WEYENBERG, HENRY,† Prod. Mgr.,

Chief Engr., Holland Furnace Co., Holland.

Minnesota

BROMMER, D. A., Estimator, H. Conrad Mfg. Co., Minneapolis.
SORENSON, R. E.,* Prod. Mgr., Htg. & A-C., McQuay, Inc., Minneapolis.

Wisconsin

AHUJA, A. K.,† Sales Engr., Vilter International, Inc., Milwaukee.
SUCHY, G. F., Inspector of Htg. & A-C., City Dept. of Bldg. & Safety Engr., Waukesha.

REGION VII

Alabama

PAYNE, JULIAN, Control Mgr., Mobile Supply Co., Mobile.
SMITH, W. J., Br. Mgr., North Brothers, Inc., Birmingham.

Kentucky

STOCKING, G. E., Engr. Mgr. Prod. Evaluation, General Electric Co., Louisville.

Louisiana

BRUNO, J. A., Pres., Brindell-Bruno, Inc., New Orleans.

Missouri

FEE, E. H., Repr., Owens-Corning Fiberglas Corp., St. Louis.
GIUDICI, G. E., Secy.-Treas., Crescent Parts & Equipment Co., Inc., St. Louis.
HENDERSON, FRANCIS, Engrg. Draftsman, Hellmer & Medved, Kansas City.
JONES, F. W., Mgr. Quotation Dept., U. S. Supply Co., Kansas City.
NEVILLE, J. W., Dist. Mgr., Bridgeport Brass Co., St. Louis.
OHLEMEYER, K. A., Test Engr., Sporlan Valve Co., St. Louis.
WOLFF, R. J., Sales Engr., Johnson Service Co., St. Louis.

Tennessee

ANDERSON, L. M., Designer & Estimator, Marr & Holman, Nashville.
AUSTIN, J. F., JR., Owner, Indoor Air Comfort, Nashville.
BOUCHARD, J. E., III, Engr., John Bouchard & Sons Co., Nashville.
BRYAN, E. E.,† Dir. Bldgs. & Grounds, Vanderbilt University, Nashville.
BURR; G. C., Jr., Engr., Memphis.
DAILEY, V. R., Vice-Pres. Mech. Sales, Boiler Supply Co., Inc., Nashville.
DEAN, R. T., Mgr., Young Insulation Co., Nashville.
DOCKUM, R. E., Vice-Pres., Service & Installation, Central Air Conditioning & Heating Inc., Nashville.
EMERY, R. O., Mfr. Repr., Nashville.
HAILEY, J. A., Gen. Mgr., Evans-Hailey Co., Nashville.
HARDISON, M. L., Sales Repr., Noland Co., Nashville.
LUNN, W. E., Pres., Boiler Supply Co. Inc., Nashville.
SUDEKUM, W. J., Sales Repr., Johnson & Scott, Hendersonville.
SMITH, W. T., Estimator, Smith Plumbing & Heating Co., Nashville.
WAECHTER, H. C., Designer, Edwin A. Keeble Assoc., Nashville.
WAGGENER, J. E., Repr., Air Conditioning Sales & Service, Nashville.

REGION VIII

Arkansas

GRIFFIN, W. L., Chief Engr., Fagan Air Conditioning Co., Inc., Little Rock.

Texas

ALEXANDER, J. B., Sales Mgr., Catlett Engineers Inc., Dallas.
CHANEY, P. E., Prod. Dvlpt. Coordinator, Texas Electric Service Co., Ft. Worth.
COBB, J. M., Customer Engr., Houston Lighting & Power Co., Houston.
LUEDECKE, W. H.*, Owner, Luedecke Engineering Co., Austin.

REGION IX

Colorado

KRIEGER, L. W., JR., Sales Engr., American Standard, Industrial Div, Denver.

North Dakota

RITCHIE, W. L., Asst. Chief Engr., American Hydrotherm Corp., Grand Forks.
RUSH, J. R.*, Chief Mech. Engr., Bison Engineering Service, Minot.

Texas

DOYLE, G. H., Repr., Southwestern Sheet Metal Works, Inc., El Paso.

REGION X

Arizona

GILBERT, JACK, Repr., Hearn Plumbing & Heating Co., Tucson.
KEARNS, R. F., Mgr., Central Air Conditioning Co., Tucson.

California

BROWN, D. R.*, Proj. Engr., Missimers Inc., Glendale.
CHRISTIAN, C. W. JR., Repr., Dow Chemical Co., Los Angeles.
CRABBE, F. J., Gen. Engr., Post Engineers U. S. Army, San Francisco.
FLEMING, R. K., Dist. Mgr., Recold Corp., Los Angeles.
HALLORAN, J. B., Mech. Design Engr., Los Angeles Water & Power Dept., Los Angeles.
KIRTLEY, D. S., Repr., Norman S. Wright & Co., San Francisco.
KONFAL, R. C.*, Sales Engr., E. L. Payne Heating Co., Beverly Hills.
KREKORIAN, HAIG, Asst. Mech. Engr., So. California Edison Co., Los Angeles.
LEWIS, R. L., JR., Sales Engr., Air Conditioning Supply Co., Los Angeles.
LYNCH, J. A.*, Mech. Engr., Gilbert Comeau, Los Angeles.
ZEILE, J. E., JR., H & V Job Leader, Ralph M. Parsons Co., Los Angeles.

Washington

LUX, R. C., Mech. Engr., John Graham & Co., Seattle.
STONE, R. C., Mfr. Repr., Seattle.
WARD, G. M.*, Facilities Engr., Boeing Airplane Co., Renton.

FOREIGN

Brazil

METH, LUIZ*, Mech. Engr., Bendix Do Brasil, Campinas, S. Paulo.

Chile

VALDES, ARTURO,† Chief Engr., Foram Chilena Ltda., Santiago.

Colombia

PINEROS, J. T., Mech. Engr., Cuellar, Serrano Gomez & Cia, Ltda, Bogota.

Guatemala

ROSETTE, RAFAEL, Chief Engr., Equipos De Bar Y Cafeteria S.A., Guatemala City, Guate.

India

CHADHA, N. S.,† Chief Engr., Mgr., C. R. Coldstorage & Ice Factory, AGRA U. P.

Mexico

BRAVO, MIGUEL, Gen. Mgr., Rapid Ice Freezing De Mexico, S.A., Mexico City.

Thailand

CHONGVATANA, PRIJA, Trainee, Vilter Manufacturing Co., Thonburi.

Venezuela

JORGE, C. N., Appl. Engr., WMCA Tecnica C. A. Caracas.

STUDENTS

JOYCE, L. J., N. Y. State Agricultural Technical Inst. Alfred University, Alfred, N. Y.
MCNAMEE, M. A., N. Y. State University Agricultural & Technical, Farmingdale, N. Y.
MISHOE, GUY H., N. Y. State Agricultural Technical Inst. Alfred University, Alfred, N. Y.
ROOS, W. G., Evansville College, Evansville, Ind.
SCHORNHORST, J. R., Evansville College, Evansville, Ind.
SIMPSON, B. R., N. Y. State Agricultural Technical Inst. Alfred University, Alfred, N. Y.
YOUNGBLOOD, W. W., Evansville College, Evansville, Ind.

With the IIR in Copenhagen

B. H. JENNINGS

During the period, August 18 through 26, the meeting of the 10th International Congress of Refrigeration was held in Copenhagen, Denmark. At this outstanding meeting 331 papers were presented by communicants from all over the world. From France there were 61 papers, the United Kingdom produced 40, the U.S.A. 35, U.S.S.R. 21, Germany 27, with lesser numbers from other countries throughout the world.

It is my feeling that the Congress was successful in almost every category, attendance, scientific import and international interchange of knowledge and good will. As would be expected, the quality of the papers varied over extremely wide limits. Some of them were merely technical descriptions of a plant or process, while others represented profound analytical developments or reported on experimentation that were true contributions to science.

Possibly one quarter of the papers might be considered in the latter categories. The benefits from such a Congress, in addition to its scientific teachings, rest on permitting researchers to learn the types of thinking and the direction in which science is moving in other countries.

In addition to the papers presented before the nine Commissions of the Congress, a number of plenary sessions were held. The first of these dealt with refrigeration machinery and among other speakers J. F. Downey Smith (U.S.A.) spoke on Trends in American Refrigeration Machinery. At the second plenary session, dedicated to Refrigerated Transport of Perishable Foodstuffs, speakers appeared from five European countries. An evening lecture was given by Dean F. G. Brickwedde (U.S.A.) On Fifty Years of Low Temperature Research and Development. The final plenary session dealt with the subject, Cold and the Human Being, with speakers from the United Kingdom and Yugoslavia.

A large number of Americans joined with those from other countries in speaking before the nine Commissions, each of which deals with different aspects of refrigeration. Because this may be of interest, the names of the Commissions are given here:

1–Scientific Problems of Low-Temperature Physics and Thermodynamics
2–Transfer of Heat, Thermal Properties of Materials and Instrumentation
3–Design, Construction and Operation of Machinery for Refrigeration and Air Conditioning
4–Applications of Refrigeration to Foodstuffs and Agricultural Produce
5–Cold Stores and Ice-making Plants
6–Applications of Refrigeration Excluding Foodstuffs and Agricultural Produce
7–Refrigerated Transport by Land and Air
8–Refrigerated Transport by Water
9–Education

At the plenary sessions, attendance reached from 200 to 600 people. Meetings of the different Commissions, which necessarily were held simultaneously, ranged in attendance from a dozen to more than 100. In fact, in a number of cases, the meeting rooms were too small to accommodate the number of those wishing to attend.

Two languages were used at the Congress, English and French, and at the plenary sessions, head phones with receiving sets were provided at each seat, so that a translator could provide a running translation in the language not employed by the speaker. In the specific Commission meetings, however, no translations were available, except that provided by bilingual chairmen or secretaries of the sessions. No serious difficulty arose as almost everyone in attendance had at least some knowledge of one of the languages of the Congress. It is unquestionably true, however, that in some of the sessions, a number of those in the audience missed some or all of a speaker's presentation.

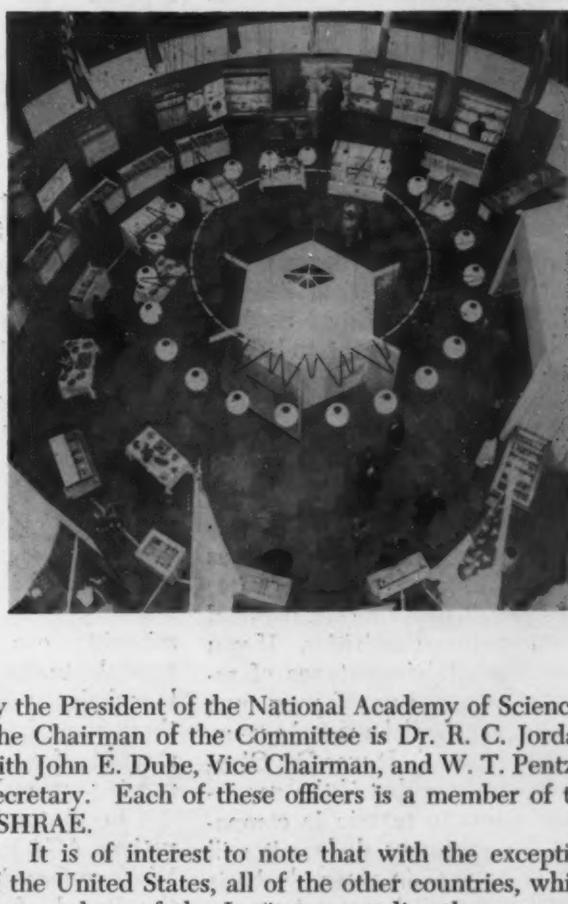
I think it has been beneficial for the United States to have become associated with the International Institute of Refrigeration, not only from the benefits obtained by those who participated in the Congress, but also because the participation of the United States has helped dissipate the atmosphere of aloofness which this country has had in a number of fields. The good will which can be created at a Congress of this type by mutual interchange of ideas is extremely valuable. Another important activity of the Institute is that of promulgating standards for refrigeration throughout the world and the Institute has already carried out some outstanding work in this connection.

The Congresses are held at four-year periods and the next Congress is scheduled for 1963, to be held in West Germany. In addition, there are meetings each year of different Commissions of the IIR. In my opinion it is of value to have engineering-scientific representatives from the U. S. attend many of these yearly Commission meetings as in this way it is possible to keep in touch with specific topic fields.

In addition to the scientific activities of the Congress, a number of business meetings were held, one

As with most of the scheduled events at the 10th International Institute of Refrigeration Congress, the Banquet at the Wivex Restaurant in Copenhagen was well attended.

This portion of the exhibits shown at the IIR Congress included frozen foods and equipment.



of these being a meeting of the Technical Board. At this meeting the officers are always selected to control technical activities until the time of the next Congress. Dr. J. C. Fidler of the United Kingdom presided and made the suggestion that not more than one presidency be allocated to any one country, and this pattern was followed in naming the presidents of the Commissions. In the subsequent nominations, the presidency of Commission No. 1 went to Professor A. van Itterbeek from Belgium; Commission 2 to Professor C. F. Kayan of the U. S. A.; Commission 3 to Professor G. Lorentzen of Norway; Commission 4 to Dr. J. Kuprianoff of Germany; Commission 5 to Mr. J. B. Verlot of France; Commission 6 to C. H. Kobulashvili of U.S.S.R.; Commission 7 to Mr. M. Fasoli of Italy; Commission 8 to K. C. Hales of the United Kingdom; and Commission 9 to T. van Hiele of the Netherlands.

Americans named to Commission Vice Presidencies were, Dr. F. G. Brickwedde, B. H. Jennings, W. T. Pentzer, and W. R. Woolrich. Mr. William Pentzer was also renamed a Vice President of the Technical Board, and Dr. R. C. Jordan was appointed a Vice President of the Executive Committee. Nominations were later approved at the General Conference which represented the formal business meeting of the Institute and at this meeting the delegates of the different countries were in attendance. In addition to these business meetings the different Commissions held their own meetings and planned programs for the years which intervene before the next Congress.

The United States is represented in the International Institute of Refrigeration by the National Academy of Sciences-National Research Council which has for its working group, the United States National Committee of the International Institute of Refrigeration. This Committee consists of 20 individuals selected from academic, governmental, and industrial institutions with each member appointed by the President of the National Academy of Sciences. The Chairman of the Committee is Dr. R. C. Jordan, with John E. Dube, Vice Chairman, and W. T. Pentzer Secretary. Each of these officers is a member of the ASHRAE.

It is of interest to note that with the exception of the United States, all of the other countries, which are members of the Institute, are directly sponsored and supported by their governments. In contrast, the United States National Committee of the IIR receives its financial support through contributions from industries, associations, and individuals who feel that this activity is a benefit both to the United States and the entire world. The United States National Committee of the IIR has received strong support from the Air-Conditioning and Refrigeration Institute, the Refrigeration Research Foundation, and from ASHRAE.

AGA seeks new horizons

41st annual meeting held in Chicago

Research needs and opportunities, as well as the rewards which may derive from their development, were hammered insistently by key speakers at the 41st Annual Convention of the American Gas Association, October 5-7 in Chicago. True, research for engineering purposes was the dominant thought expressed but that related to industry growth, to service and to marketing and sales received strong emphasis. Indeed, the theme of the meeting was New Horizons and that was projected beyond gas technology to legislation, finance, regulation, supplies, future markets and similar management and operating problems.

Dr. T. Keith Glennan, Administrator, National Aeronautics and Space Administration spoke with conviction in regard to technical research in industry ". . . science and technology have emerged as essential ingredients in all phases of our personal, corporate, national and international activities. If you ignore the vital importance of research to the future of your own industry, you will have only yourselves to blame." It was Dr. Glennan's contention that if the gas industry wants to remain in competition for customer dollars it will have to engage in research—basic as well as applied research—to an extent that he feels few gas company executives seem to have been willing to contemplate.

The speaker viewed the most precious ingredient in any research effort as the continuing supply of well educated, highly motivated men and women and asserted that to give them the best chance for success in their endeavors it was necessary that they be assured a stable and adequate level of support both in the financial sense and in the confidence in their efforts which management must have and express. "In other words," maintained Dr. Glennan, "a favorable climate must be created and consciously maintained if the research effort is to be fruitful."

As to how to go about it, the speaker called for first, a conviction on the part of top management that research is a necessary and vital part of corporate activity not a luxury where "long-haired scientists and engineers dissatisfied with the present state of the art indulge themselves" and second, that the program should include a substantial amount of basic scientific research either within a company's own laboratories or through generous and unfettered support of graduate and research programs on the campuses of institutions of higher learning. "Actually, before the scientists can undertake fundamental studies in search of new knowledge—real basic research—there must be done an enormous amount of creative work in engineering research and development."

In substantiation of these observations, AGA President J. Theodore Wolfe referred to the decision of the Board of Directors for a research budget of \$2½ million for 1960 to maintain the strong position of the gas industry in the "battle of the fuels." President Wolfe anticipated the raising of the level of expenditure for cooperative research by AGA to \$6 million by 1965. One of the expectations, or goals, of the gas industry for 1970 is 45 million customers, a doubling of gas sales volume and a gross plant investment 2½ times what it is today.

Upgrading of gas appliance quality was the plea of President Edward A. Norman of the Gas Appliance Manufacturers Association who addressed the general session on October 5 saying, "Events of 1959 already bear out the contention of many manufacturers that the public will accept and pay for quality when and where . . . the market is conditioned to it." Speaker Norman placed emphasis upon merchandising factors but stressed that "the industry is becoming more and more research-minded, more and more capable of interpreting its findings and readier than ever to produce what these findings show it must produce to meet the demands of the 1960's." Among the cited signs of the times were such developments as a two-furnace and two-boiler home market, year round gas air conditioning and a vastly improved gas refrigerator outlook. He expressed satisfaction in regard to housing developments for senior citizens which are tending toward being equipped largely for gas.

Carrying the basic theme one step further, Dr. J. T. Rettaliata, President of the Illinois Institute of Technology, referred in some detail to the work of the Institute of Gas Technology, to the places being taken by alumni of that Institute at various points in industry and to the need for continued attention to our educational system, its output of competent and alert engineers and the opportunities which may be afforded them within the gas industry.

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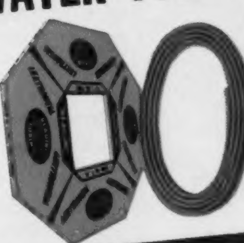

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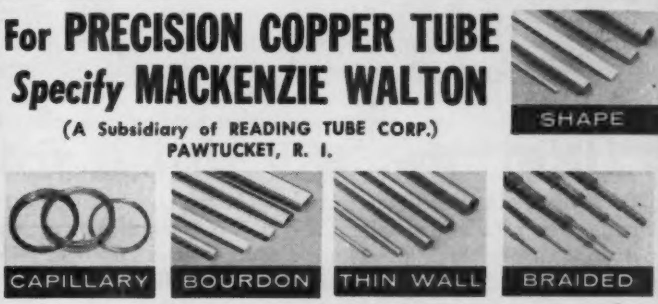
A line of heat transfer tubing made by Readi-Fin Mfg. Co., Inc. (a subsidiary of Reading Tube Corp., at Reading, Pa.). The extended surface is extruded from the tube wall, thus eliminating the possibility of fin failures resulting from thermal shock, corrosion and erosion. This "one-piece" (integral) construction provides maximum heat transfer efficiency, ruggedness, easy fabrication and freedom from fouling. It's available in Water Tube types with finned or plain ends and in Condenser Tube types with finned, plain or stripped ends.

For PRECISION COPPER TUBE

Specify **MACKENZIE WALTON**

(A Subsidiary of READING TUBE CORP.)
PAWTUCKET, R. I.



- Famous for Accurate and Fine Finish Tubing
- Quality Controlled from Start to Finish

Sold Through Wholesalers Only ®

MADE IN U.S.A. TO THE STANDARDS OF AMERICAN INDUSTRY

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BULLETINS and CATALOGS

Solar Screens. Six basic forms of Sol-R-Wal solar screens may be used to form a wide variety of patterns for walls, panels, space dividers, privacy screens, garden walls and equipment enclosures. Aiding in reduction of heat gain through windows, these screens are effective in lowering load on air conditioning equipment. Bulletin No. MFL-5559-138, 4 pages.
Malvern Flue Lining, Inc., Malvern, Ohio.

Plastics Pipe. Noted for its impact strength and durability, Ace-It, a general purpose rigid plastics pipe for processing industries, utilities, irrigation, mining, farming, marine and general industrial use, is described in 16-page Bulletin No. CE-80. A chemical-resistant Type I ABS material, it is non-toxic, odorless and tasteless. Technical data in the bulletin covers fittings and valves as well as pipe.
American Hard Rubber Company, Div of Amerace Corporation, Ace Rd., Butler, N. J.

System-Engineered Components. Capacities, dimensions, heat transfer data and other necessary engineering information are presented in 8-page Catalog No. 300B on shell and tube condensers, shell and coil condensers, small capacity water chillers, heat exchangers, oil separators, liquid receivers and vertical shell and coil condensers.
Acme Industries, Inc., 600 N. Mechanic St., Jackson, Mich.

Tubeaxial Air Unit. Included in Flyer 5912, descriptive of Air Unit No. 89B222 for ventilating electronic equipment, are installation diagrams giving major assembly dimensions, performance data and a curve showing rated delivery at various static pressures.
American Radiator and Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

Thermometers and Regulators. Four catalogs present information on this manufacturer's instruments: Catalog 18-A on Mercury Actuated Thermoregulators and Thermostats, Catalog 19-B on Thermometers and Hydrometers for Laboratories and Plants, Catalog 20-A on Home, Farm and Utility Thermometers, and Catalog 21 on Psychrometers, Midget Industrial Thermometers and Dial Thermometers. Products are illustrated.
H-B Instrument Company, American and Bristol Sts., Philadelphia 40, Pa.

Remote Room Conditioners. Information in 4-page Catalog No. 381A covers vertical and horizontal units with and without cabinets, for this line of individual room air conditioners for multi-room buildings. Four sizes are available, 200, 300, 400 and 600 cfm, in the four types of conditioners offered, manufactured for use with both heating and cooling coils. Dimensions, specifications and capacity data are given.
Acme Industries, Inc., 600 N. Mechanic St., Jackson, Mich.

Back Pressure Regulator Valve. Pilot-operated and suitable for pressures to 250 psi wsp and temperatures to 500 F, this regulator is available in ½- to 2-in. sizes with screwed ends and 2½- to 3-in. sizes with flanged ends. Flyer JNP-3 outlines features and materials of construction, giving recommended capacity and outlet pressure range.
OPW-Jordan, 6013 Wiehe Rd., Cincinnati 13, Ohio.

Temperature Regulators. Depending upon the application, these regulators, for use on water heaters, fuel oil preheaters, fuel oil coolers and storage tanks, can be supplied as either direct-acting, for heating applications, or reverse-acting, for cooling applications. Vapor-pressure operated, they are designed to control the flow of either heating or cooling fluids. Four-page Bulletin No. 7-1 fully describes the Type TC and Type TCX Series, including capacity information.
A. W. Cash Valve Manufacturing Corporation, 666 E. Wabash Ave., Decatur, Ill.

Induced-Draft Fans. Two publications, 4-page Bulletin L-1A and 16-page Bulletin L-3, combine to provide dimensional data, recommended sizes, material specifications, suggested stack sizes, various outlet positions, and rating tables of these fans, based on 600 F at sea level.
Lehigh Fan and Blower Div, Fuller Company, Catasauqua, Pa.

Copper Tube Fittings. Containing much information on the making of solder joints and flared connections, this fitting bulletin, 44-page Catalog SF-59, also includes data on types of solder and working pressures, dimensions of copper water tube, and flow capacities and friction loss allowances for both tube and fittings. Covered are wrought and cast fittings for copper water tube and drainage tube in both flared and solder-joint types. Threaded, solder-joint and flared valves available are also listed, as are the recommended and approved accessories, such as solder, straps, fluxes and sizing and flanging tools.
Chase Brass and Copper Company, Waterbury 20, Conn.

Draft Inducer. Mechanically powered and intended to provide additional draft in chimneys when natural draft is not sufficient, the Shur-Flo Draft Inducer is cited as correcting most draft problems in fuel-burning heating equipment. 16-page Bulletin SF-59, "How to Cure Sick Chimneys," describes causes of chimney failures, suggests means for selection of a draft inducer and gives capacities and performances of this line.
Walker Manufacturing and Sales Corporation, Dept. S-F, St. Joseph, Mo.

Gasket Design Manual. Contents of 8-page Form 359 cover product description, applications, materials and types of rings available, means of selection and how to order Metal O-Ring gaskets. Included are four slip-in pages detailing size and groove dimensions.
Advanced Products Company, 59 Broadway, North Haven, Conn.

Flexible Cushion Couplings. Ranging from fractional to 190 hp per 100 rpm, an expanded line of Para-flex couplings is described in 12-page Bulletin A669C. Two new sizes have been added to the 10 previously offered to increase the scope of industrial applications for which the coupling can be used. PX24, largest in the line, is capable of delivering up to 2000 hp at 1080 rpm, 2½ times the maximum hp of the previous largest coupling. Cuts and text explain design features and operating characteristics of the line.
Dodge Manufacturing Corporation, Mishawaka, Ind.

Bulk Milk Coolers. Three lines, "Low Line," "Space Saver" and "Big Standard," consisting of five, three and four models, respectively, are presented in four-page Bulletin 209. Front and side elevations are shown here for each line and dimensions and capacities are given.
Sunset Equipment Company, P. O. Box 3536, St. Paul, Minn.

BULLETINS

Resistance Elements. Available in a wide variety of sizes, shapes and materials, RdF Stikons consist of a thin sandwich of two sheets of insulation with a temperature-sensitive wire grid bonded between them. Exceedingly fine wires are drawn from nickel, platinum or nickel-iron alloys, wound into grid forms and provided with suitable lead wires for trouble-free electrical connections. In size and shape the element is comparable to a postage stamp. Six-page Bulletin T59 brings readers up-to-date on all new designations and data for the elements which, when cemented to a surface, measure temperature within range to a fraction of a deg.
Arthur C. Ruge Associates, Inc., Hudson, N. H.

Expandable Polystyrene Insulation. Charts and cuts illustrate the various insulating uses of this plastics material in home freezers, refrigerators, portable coolers, trucks and air conditioning lines, as well as for wall, ceiling and floor material in freezer rooms. Detailed in the charts are comparisons between Dylite and other conventional insulating materials. Typical properties are listed in six-page Bulletin C-9-271 and advantages of the product are cited with respect to the K factor and variations of thermal conductivity with temperature and density.
Koppers Company, Inc., Plastics Div, 801 Koppers Bldg., Pittsburgh 19, Pa.

Packaged Automatic Boilers. Describing and illustrating boiler types, including gas, oil and combination gas-oil models; a specially-designed hot water boiler; and the steam atomizing principle for use with No. 6 oil, this 12-page bulletin includes ratings and dimensions of all sizes in the line.
Orr & Sembower, Inc., Reading, Pa.

Testing Facilities and Service. Describing this company's Materials Evaluations Div's facilities and services for analysis, development, research and inspection of materials and products, is 6-page Bulletin 5902. Among the services offered are metallurgical studies, metals chemistry, plastics evaluation and physical testing, done with the aid of a complete range of analytical equipment such as an x-ray diffraction unit, emission spectograph, spectrophotometer, a variety of tensile and compression test units and heat treating furnaces.
United States Testing Company, 1415 Park Ave., Hoboken, N. J.

Four-Way Valve. Of all metal construction, the ⅞-in. valve provides four-way shifting with a conventional three-way control, eliminating midshift stalls and allowing a greater range of operating pressures. Features of the valve are detailed in a 4-page bulletin.
Product Engineering, 9703 Southwest Hwy., Oak Lawn, Ill.

Gearmotors. Available from ¼ to 100 hp, the gearmotors covered in 8-page Bulletin 51B9172 are integral and all-motor types, including right angle as well as special designs. Construction and design features are given.
Allis-Chalmers Manufacturing Company, Milwaukee 1, Wisc.

Packaged Refrigeration Units. Pre-engineered for freezing, cooling and ice-holding systems, these units are completely air-cooled and require no additional plumbing. Easily installed, they are fitted through a wall opening of the suitable size, secured and plugged into a suitable electrical outlet. Four-page Bulletin SP-5/59 10M

presents capacity information and specifications for all four series. Product features are detailed.
Taylor-Burch Refrigeration Products, Inc., 2524 Brooklyn Rd., Jackson, Mich.

Differential Pressure Transmitters. Inline mounted flow rate transmitters for low-flow metering of either liquids or gases, Model 10B1460C Series instruments utilize a differential pressure transmitter, shunt tube and an interchangeable nozzle assembly. Transmitter output of 3 to 15 psig is linear with differential pressure and can be used to indicate or record flow rate on a square root scale with conventional pneumatic receivers. Four page Specification 10B1460 covers materials of construction, performance and operational limits and includes sizing procedure and a sizing guide.
Fischer and Porter Company, Hatboro, Pa.

Chilled-Water Cooling Coils. Available in a wide range of sizes for central station or zone air conditioning systems, these coils feature a fin with an embossed pattern cited as increasing fin surface and causing better air wash, resulting in up to 15% greater heat transfer. Flyer CWC-202 gives available coil sizes, dimensional data and standard ratings.
Halstead and Mitchell, Bessemer Bldg., Pittsburgh 22, Pa.

Subminiature Elapsed Time Indicator. Incorporating jewel bearings, a 1.8 billion to one gear train and a low inertia synchronous motor, the WT-1 subminiature elapsed time indicator weighs under 3 oz and has a total read-out of 10,000 hr, readable to the closest hour. Flyer 5001 describes the physical and electrical characteristics of the instrument and includes dimension drawings and an actual size template for panel layout.
Waltham Precision Instrument Company, 221 Crescent St., Waltham 54, Mass.

Air Conditioners. Self-contained air conditioners and an air-to-air remote heat pump system are the subjects of Flyers PA-1 and 371-59, respectively. Consisting of two parts, a remote compressor outdoor coil unit and a universal blower coil indoor unit, the year-round air conditioner heats in winter without fuel and cools in summer without water, using only electricity and air. Included is information on performance and tables giving dimensions and specifications.
Heat Controller, Inc., Jackson, Mich.

Oil and Gas Burner Combination. Design of this mechanical pressure atomizing oil and nozzle-mix gas burner is explained in a 6-page folder showing construction features and principle of operation.
Boiler Engineering and Supply Company, Inc., Phoenixville, Pa.

Temperature Regulators. Depending upon the application, these regulators, for use on water heaters, fuel oil preheaters, fuel oil coolers and storage tanks, can be supplied as either direct-acting, for heating applications, or reverse-acting, for cooling applications. Vapor-pressure operated, they are designed to control the flow of either heating or cooling fluids. Four-page Bulletin No. 7-1 fully describes the Type TC and Type TCX Series, including capacity information.
A. W. Cash Valve Manufacturing Corporation, 666 E. Wabash Ave., Decatur, Ill.

Centrifugal Pumps. Bronze and iron centrifugal pumps with capacities up to 320 gpm and pressures to 40 psi are the subject of 6-page Bulletin 1004, which contains a non-technical description of design features of the

Western Electric Company

NEW MERRIMACK VALLEY WORKS

32 Acres of Floor Space

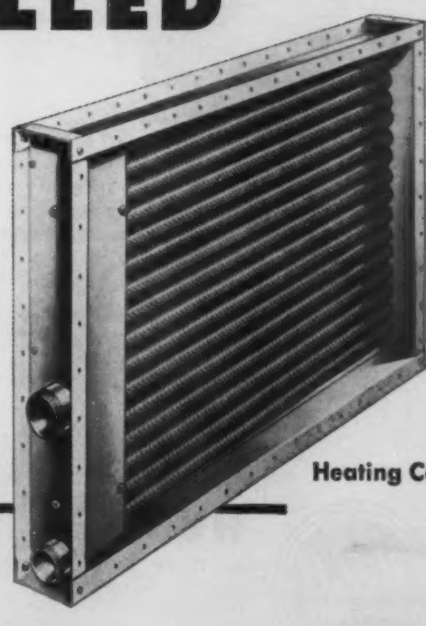
Charles T. Main, Inc., Architects-Engineers
Turner Construction Co., General Contractors, Original Plant
Perini Corporation, General Contractors, Plant Extension
M. J. Flaherty Co., Mechanical Contractors, Original Plant
John Ventura Company, Mechanical Contractors, Plant Extension

Cooling Coils

AEROFIN INSTALLED

Modern smooth-fin design of Aerofin coils permits ample heat-exchange capacity in limited space—permits the use of high air velocities without turbulence or excessive resistance.

Aerofin performance data are laboratory and field proved. You can safely specify Aerofin coils at full published ratings.



Heating Coils

AEROFIN CORPORATION

101 Greenway Ave., Syracuse 3, N. Y.

*Aerofin is sold only by manufacturers of fan system apparatus.
List on request.*

ENGINEERING OFFICES IN PRINCIPAL CITIES

pumps. Performance data on clockwise, counter-clockwise and reversible pumps are included, together with specifications and reasons for claims of high efficiency in operation.
Marine Products Company, 670 Lycaste Ave., Detroit 14, Mich.

Purging Refrigeration Systems. Data on automatic removal of air from refrigeration systems is provided by 12-page Bulletin No. 700, cited as answering such questions as why purging is necessary, how non-condensible gases affect efficiency, methods of purging, where to make purge connections and how to use the refrigerated purger.
Armstrong Machine Works, Three Rivers, Mich.

Roof-Mount Heating and Cooling. Intended for single-story buildings of all sizes, roof-mount units are placed directly above the area to be conditioned, with no ducts required. Individual thermostats control each unit. Included in a six-page bulletin are condensed specifications and description of operating features.
Ventil-Aire Corporation, 1815-17 Decatur St., Brooklyn 27, N. Y.

Rubber Parts. Categorizing available molded and extruded rubber parts in six groups, 6-page Catalog AD-167 offers a detailed description of each type, along with typical applications. In order to aid the user in choosing the right material for his specific application, the bulletin has a page of tabular material showing the tensile strength, elongation and compression for natural, SBR, nitrile, neoprene, Butyl and specialty fluoroelastomers Viton and Kel F. A second page of tabular material compares the physical properties and effect of environments on these types.
Garlock Packing Company, 436 Main St., Palmyra, N. Y.

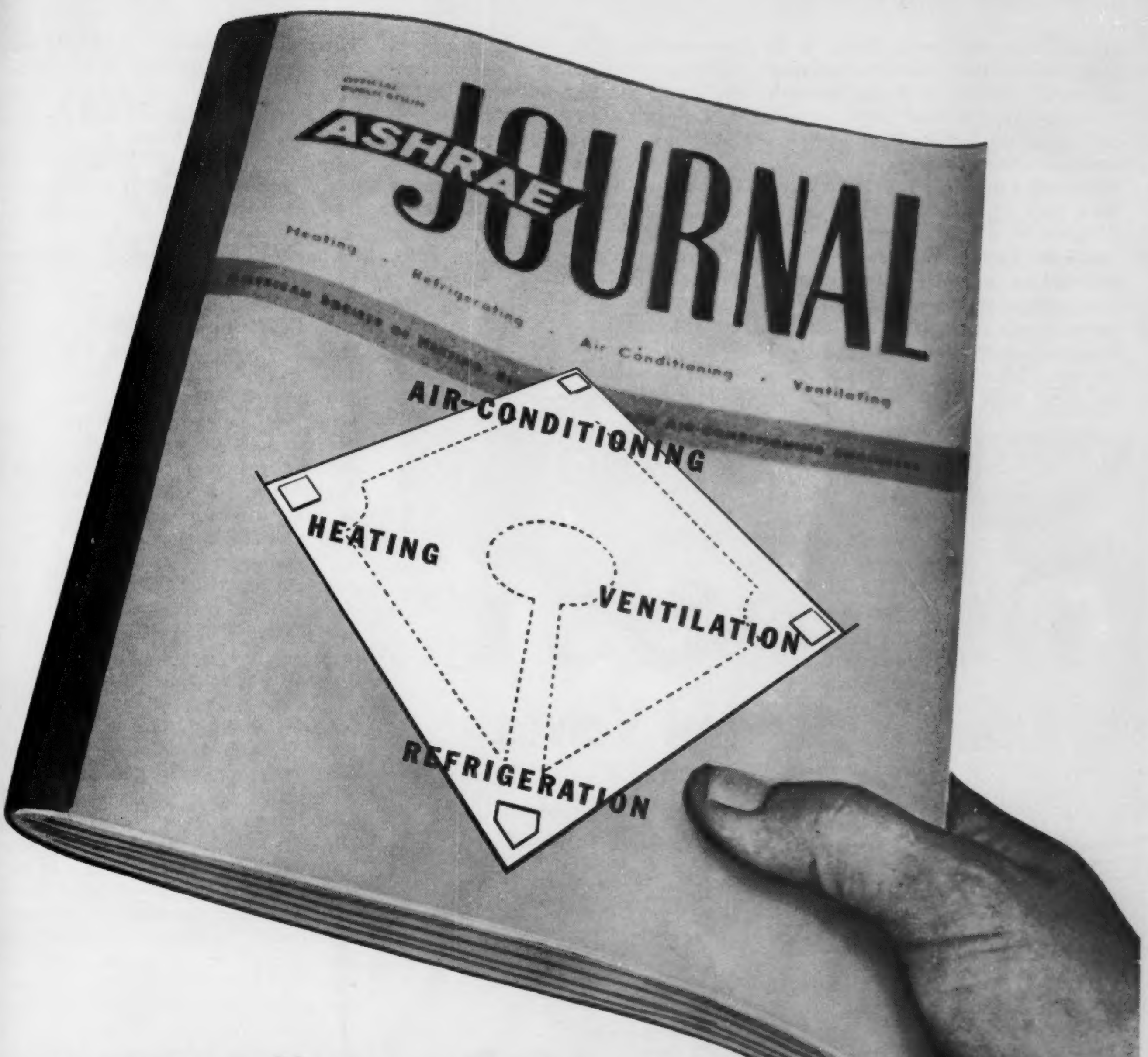
Packaged Automatic Boilers. Recommended for installation in apartments, churches, smaller schools, hospitals, warehouses and for low-pressure steam supply in many industrial and food processing applications, Power-Pak boilers are available in the 15 to 25 hp range. Designed for low-pressure steam or hot water heating, they include integral oil, gas or combination firing equipment and a wide selection of control and safety features. A 4-page bulletin offers descriptions.
Orr and Sembower, Inc., Reading, Pa.

Refrigerant Filters. Procedure for cleaning a refrigeration system after hermetic burnout and low side filtration are discussed in 6-page Bulletin A-12, which also gives ratings and specifications for Permaclean liquid and suction refrigerant filters.
McIntire Company, Livingston, N. J.

Steam Coils. Tables covering circuiting selection, steam coil selection method, capacity at common steam pressures, dimensions, face areas of coils, and air friction are included in Catalog 51C9a. Design changes are incorporated, as well as illustrations showing circuiting of the coils to meet the individual needs of the job and minimize stratification.
Recold Corporation, 7250 E. Slauson Ave., Los Angeles 22, Calif.

Duct Fans. Belt-drive, direct-drive, bi-pass and reversible duct fans are included in 12-page Bulletin A-114A, which gives product cuts, principal dimensions and ratings for the line.
Hartzell Propeller Fan Company, Div of Castle Hills Corporation, Piqua, Ohio.

Iron Body Gate Valves. Listing service ratings, sizes and dimensions for this series of valves, applicable to many industries, including the construction, gas transmission, chemical,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electric and petroleum fields, is 6-page Bulletin 10M-7-59-180. Included is a section devoted to Higher Strength Cast Iron Wedge Gate Valves, which are used on many lines carrying steam, water, oil and gas.
Walworth Company, 750 Third Ave., New York 17, N. Y.

Gas Line Testing Device. Designed to build up and maintain a constant test pressure in low and intermediate pressure gas lines, the Leak Seeker is a small, electrically operated compressor. Self-contained, the unit can be used with any of the type of manometers or gauges now utilized for testing lines, and operates on regular house current. Bulletin F-332-A, four pages.
DeVilbiss Company, Toledo 1, Ohio.

Half-Round Unit Cooler. Air drawn in through a fan is discharged radially in a 180 deg arc for maximum air distribution in HR cooling units. Physical data, capacity data and a direct selection table for matching the cooler with the condensing unit are given in Flyer 36.
Bohn Aluminum and Brass Corporation, Betz Div, Danville, Ill.

Temperature Controls. Used both for controlling and as high or low temperature safety alarms and cut-outs, these devices are available for temperatures from sub-zero to 2000 F. Illustrations of all standard models, brief description of method of operation and specifications are included in 4-page Catalog G-22.
Burling Instrument Company, 16 River Rd., Chatham, N. J.

Valved Couplings. Designed for "no-spill" service in accordance with military specifications for use in airborne and ground hydraulic systems, 15 Series couplings are within specification requirements for spillage, air inclusion, pressure drop and envelope. Available in ¼, ⅜, ½, ¾, 1 and 1¼-in. sizes, the couplings may be used with fuels and other fluids with working pressure up to 3000 psi and temperatures up to 400 F in the ¼-in. size. Product cuts, dimensional data and selections of end threadings are included in 4-page Catalog 280B.
Snap-Tite, Inc., Union City, Pa.

Silicone Potting Material. Curing in place to form a resilient, protective mass which retains its dielectric properties and moisture resistance over a temperature range of –60 to 200 C, Dielectric Gel is a silicone potting material that permits visual and instrument checking of individual parts within a potted assembly. In addition to tables of properties, 4-page Bulletin 10-505 includes a graph cited as showing that no damaging stresses are exerted on delicate parts by this material either during or after cure.
Dow Corning Corporation, Midland, Mich.

Pipe and Fin Coils. Prepared for the design engineer, the 1959 Edition of this Engineering Data Book contains 48 pages of tabulated data giving heat transfer coefficients for all heating and cooling problems involving the application of pipe or fin coils, cuts illustrating various coils, bends and fabricated piping and coil design specification sheets. A feature of this edition is the inclusion of depressant factors for pipe and fin coils when operating at high altitude conditions such as encountered in altitude assimilation chambers.
Rempe Company, 340 N. Sacramento Blvd., Chicago 12, Ill.

Dielectric Pipe Unions. Acting as an electrical insulator when connected in plumbing systems to dissimilar metals, these unions control corrosion caused by galvanic or electrolytic action.

(*Continued on page* 114)

BULLETINS

(*Continued from page* 110)

Eight-page Catalog DU 19 includes technical specifications and performance data for the device, as well as information on the possible results of contact between dissimilar metals.
Epco Sales, Inc., 3204 Sackett Ave., Cleveland 9, Ohio.

Main Service Equipment. Providing four fusible units which may be used for such heavy duty appliances as air conditioners, the 100 amp Renu-Fuse device, for which complete specifications and wiring diagrams are given in Flyer 49A, controls all circuits.
Wadsworth Electric Manufacturing Company, Inc., Covington, Ky.

Permanent Furnace Filters. Featuring an all-aluminum frame and non-corrosive construction plus a plastic filter that removes dirt, dust and impurities from the air, the Sa-Foam Filter requires no oil and is care-free, cleaned by flushing under running water. Flyer SG-T-59.
Safeguard Corporation, Lansdale, Pa.

Water Treatment. Formulated to prevent scale, corrosion and slime formation in cooling towers and evaporative condensers, EZ Treat is a complete, one package treatment. No special feeding apparatus is necessary, nor is testing. Method of use is detailed in a flyer.
Metropolitan Refining Company, Inc., 50-23 23rd St., Long Island City 1, New York.

Industrial Insulation. Listed in 4-page Bulletin 6451 are all types of industrial insulations and insulating cements. Indicating applications for high temperature and superheated surfaces; high, medium and low pressure steam lines; cold water pipes; heating and air conditioning ducts; and breechings is a tabular product selection guide.
Philip Carey Manufacturing Company, 320 S. Wayne Ave., Lockland, Cincinnati 15, Ohio.

Cooling Towers. Lighter operating weight, closed distribution system cited as preventing formation of algae and keeping dirt and litter from blowing in to clog the distributors, completely sealed motor and canted collecting pans which wash debris into the depressed sump characterize the Keystone Lo-Line Cooling Towers, capacity tables and suggested specifications for which are detailed in 4-page Bulletin 51-902.
J. F. Pritchard and Company of California, 4625 Roanoke Pkwy., Kansas City 12, Mo.

Boiler Water Control System. In addition to the boiler water control functions of this new system, 6-page Bulletin 34-B-1 reports performance tests cited as showing how the unit can save almost 20% on fuel consumption, 75% on treatment chemicals, 75% on blow-down heat and water losses and 100% on contaminated condensate losses. Features, flow diagrams, analysis tables, specifications, dimensions, capacities and controls are also presented.
Sparkler-Filtrion Corporation, North Chicago, Ill.

Hood Ventilators. Detailed description of this line of packaged range hood ventilators for kitchens in mass-feeding establishments is given in 8-page Catalog 1201. A self-contained unit complete with grease filters, blower, motor drive and cabinet, the Filtaire eliminates grease, smoke, heat and odors by blower pressure directly above the range. Contents of the booklet include: necessity for grease filters, advantages, specifications and dimensions for basic

models and special units for high static application.
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, Ohio.

Panel Meters and Pyrometers. Indicating panel meters and pyrometers in numerous sensitivities, styles and sizes, for monitoring any electrically measurable variable, are described in 12-page Bulletin 107. General specifications of meters are listed and then grouped according to case types.
Assembly Products, Inc., Chesterland, Ohio.

Ice Cream Holdover Cabinet. Main feature of this truck cabinet is a separate compressor unit that may be used remotely or mounted at numerous heights on the back or either end of the cabinet to accommodate every sort of installation. Flyer 2213 6M 3-59 gives features and specifications of Model LTT-024.
Brewer-Titchener Corporation, Refrigeration Div, New Milford, Pa.

Pumping Equipment. Commemorating its 50th anniversary, this company has published a comprehensive 8-page Bulletin (6-59) identified as "50 Dynamic Years Serving the Nation." Diagrammatically illustrated are new products, systems and equipment.
Chicago Pump Company, 622 Diversey Pkwy., Chicago 14, Ill.

Flame Burners. Compact blast-type inshot burners, especially designed for incinerators, furnaces, boilers and other miscellaneous heating applications, Incino-Flame Burners, as described in 4-page Data Sheet 8E-1 and Flyer 8E-2, have a wide range of capacities. Models IF 450 and IF 600 have capacities ranging from 50,000 to 450,000 and 600,000 Btu/hr, respectively, and Models IF 750-E, IF 1000-E and IF 1500-E capacities ranging from 400,000 to 750,000, 1,000,000 and 1,500,000 Btu/hr, respectively.
Bryant Industrial Products Corporation, 17700 Miles Ave., Cleveland 28, Ohio.

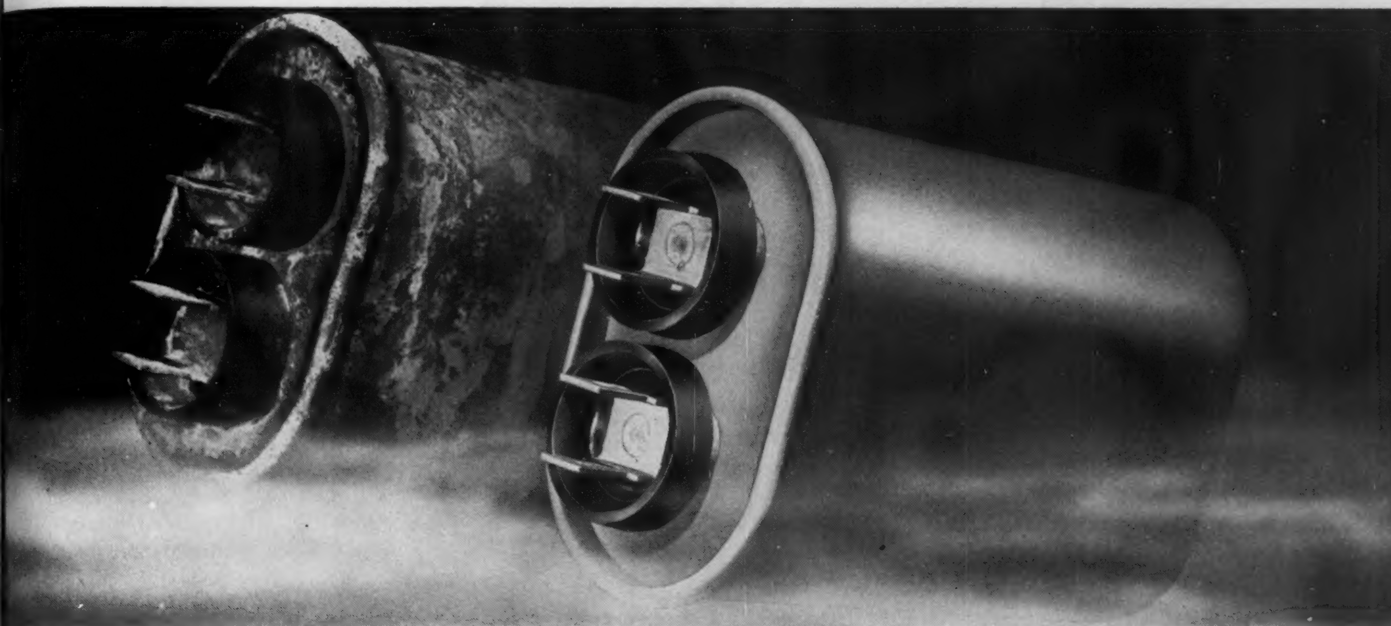
Unit Cooler. Designed with a low silhouette so that it can be mounted in the top of a refrigerator, making the entire top shelf area useable, is the Top-Aire unit cooler. Desirable for display-type refrigerators, it can also be used in back bars, under counter cabinets or wherever space is at a premium. Flyer 80 gives physical and capacity data.
Bohn Aluminum and Brass Corporation, Betz Div, Danville, Ill.

Vinyl Spray Coating. Covering MVC-1 vinyl spray in aerosol containers, this folder lists major features and benefits of the vinyl-coating, effective in sealing-out rust and corrosion in metal surfaces.
H. K. Porter Company, Inc., National Electric Div, Porter Bldg., Pittsburgh 19, Pa.

Welding Equipment. Featuring a full line of gas welding, cutting and allied equipment is a 40-page manual with 6-page insert of product cuts. In addition to equipment description, the bulletin carries charts and technical data on the various items. A section is reserved to flow capacity curves for this organization's line of gas pressure regulators.
Smith Welding Equipment Corporation, 2633 Fourth St., S. E., Minneapolis 14, Minn.

Gas Air Conditioner. Offered by this manufacturer is Bulletin 759 FQ on a recently-introduced gas air conditioner, the Oasis. An air-cooled remote design, the unit has coils which can be added on to existing forced-air heating units or a fan coil unit for

G-E Drawn-case Capacitors Now Offer Greater Protection Against Corrosion



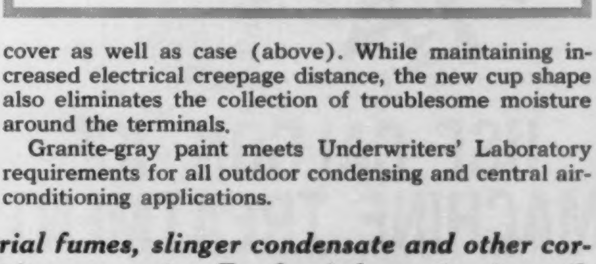
Unique Bushing Design and Automated Application of Granite-gray Paint Assure Uniform, Durable Finish

Accelerated tests in simulated industrial atmospheres proved that this new G-E Granite-gray paint finish inhibited rust and stain formation many times better than finishes used earlier.

Where other finishes were heavily corroded in as little as 250 hours in a 20% salt fog at 95F and 85-90% relative humidity, the Granite-gray paint showed only minor effects, even after 1000 hours.

Chosen only after comparative evaluation of 46 different finish systems under adverse climatic conditions, this remarkable new finish results in longer, trouble-free capacitor life.

To assure you of the maximum protection that Granite-gray paint offers, it must be applied in a uniform film completely covering the capacitor case. General Electric assures complete uniform coverage by use of a unique beveled phenolic-cup bushing and fully automated application of the paint. The new beveled cup facilitates complete paint protection of cover as well as case (above). While maintaining increased electrical creepage distance, the new cup shape also eliminates the collection of troublesome moisture around the terminals.



NEW BEVELED-CUP BUSHING

Granite-gray paint meets Underwriters' Laboratory requirements for all outdoor condensing and central air-conditioning applications.

For maximum protection against salt fog, industrial fumes, slinger condensate and other corrosive elements, specify G-E Capacitors with Granite-gray paint. Further information is available from your nearest G-E Sales Office, or write General Electric Company, Ft. Edward, N. Y.

448-6

GENERAL ELECTRIC

use when the installation is for an independent cooling system. Included in the 6-page bulletin is a full page of graphic installation diagrams and complete product information and specifications.
Day and Night Manufacturing Company, P. O. Box 2222, La Puente, Calif.

Blower Units. Complete dimensional and performance specifications on two series of direct-drive blower units with either Center Lock or standard double Airotor wheels are given in two 4-page Bulletins, DD-154 and DD-173. Performance curves give variations in static pressure, speed, power, current and air flow rate for the four blower models in each of two wheel diameter sizes. Different sets of test curves are given for units running on a number of the most commonly used commerical blower motors.
Torrington Manufacturing Company, Air Impeller Div, Torrington, Conn.

Steel Tubing. Smoothweld tubing is soldered or silver brazed without the need of separate couplings to provide one-piece systems, is easy to install in any area and has a smooth internal surface for minimum pressure drop. A 4-page bulletin discusses the application of this steel tubing in replacing screwed or welded piping.
Standard Tube Company, 24400 Plymouth Rd., Detroit 39, Mich.

Chemical Treatment for Cooling System Water. Explained in this flyer is the method by which the Chemistat, a chemical feeder, aids in reducing damage caused by the circulating water. Supplying the feeder as part of a plan, this company provides for analysis of water samples, keeping of treatment control records and regular reports on treatment.
Water Service Laboratories, Inc., 423 W. 126th St., New York 27, N. Y.

Plastics Insulation Board. Intended for perimeter insulation, Scorboard is a rigid plastics form board that can be used for foundation and cavity wall insulation as well, in both residential and commercial construction. Thermal and physical properties, product specifications and installation procedure are included in 4-page Bulletin 157-43-58.
Dow Chemical Company, Midland, Mich.

Test Chambers and Refrigeration Equipment. Including process equipment for liquid temperature control and air and metal processing, in addition to environmental test chambers, is 4-page Bulletin G-58. Ranges for temperature testing chambers are from –300 F (with liquid nitrogen) to 1000 F, full or limited range. Conditioning systems are matched for individual customer requirements, and heating is directly or indirectly by electrical resistance heaters.
Missimers, Inc., 3206 Los Feliz Blvd., Los Angeles 39, Calif.

Wall-to-Wall Convectors. For schools, churches, commercial, public and institutional buildings, Perma-Trim Wall-to-Wall Convectors were designed primarily for use on outside walls beneath large expanses of window area. By blanketing the glass with an uninterrupted curtain of warm air, these convectors replace heat loss and provide uniform, draft-free, floor-to-ceiling heating. Specifications are given in 12-page Bulletin 259-A.
Modine Manufacturing Company, Racine, Wisc.

Branch Connection Fittings. Describing the new line of fittings of this manufacturer is 6-page Bulletin TF-1-59, which illustrates the TeeLet and Fishmouth models. All pertinent design information, sizes, material specifica-

(*Continued on page* 126)

Electrowheel*®

another LAU "first in the industry"

Here's another Lau first, designed to help solve those cramped space blower installation problems. The versatile Lau "Electrowheel" is recommended for use whenever air moving efficiency is a requirement but space limitations present a problem. The "Electrowheel" is extremely efficient when operating where the utmost in compactness and smooth, quiet operation is required.

Lau "Electrowheel" features include stationary rubber mounted shaft, sealed ball bearings, rigid one piece motor mounts, 30" motor leads with BX connector and the same high standard of quality found in every Lau engineered product. One moving part assures years of trouble-free service.

When an installation requires a high performance blower the Lau "Electrowheel" is the logical answer. Write for Lau Catalog LSO-463 for complete information.

So named because a high quality external-rotor motor forms the wheel hub . . gives you more air delivery in a small package than ever before possible!

THE LAU BLOWER COMPANY, 2027 Home Avenue, Dayton 7, Ohio

Applications

SUBMARINE AIR CONDITIONING

Cooling and heating for Disneyland's fleet of eight submarines is provided by Recold air conditioning and heating coils. Part of the system includes diffusers placed in frames of the individual passenger view ports located below the water line, to prevent moisture accumulation on the inside.

One split circuit coil for direct expansion is used in conjunction with two shell-and-tube condensers, two two-ton compressors and one hot water coil. Coils are located in the conning tower of the submarine and condensers and compressors are in the machinery section at the rear. Air is recirculated through an activated carbon filter to eliminate odors and a replaceable media filter to eliminate dust. Since the craft is never completely under water, fresh air is vented into the coils through louvers at the front and top of the conning tower.

RADIANT HEATING USED IN HOSPITAL

Except for a few baseboard areas, the four-story treatment building of Connecticut State Hospital, including basement, is radiant-heated with 93,000 ft of one-in. diameter Byers corrosion resistant wrought iron pipe coils. The installation was given a weld soap test for 12 hr under 125 psi before being buried in concrete flooring. Operating from either a room air thermostat or an outdoor bulb, the system uses hot water, pumped by an Aurora circulator with a capacity of 200 to 270 gpm against a 30-ft head, as the heating medium. The 16-zone radiant heating system was designed to meet minimum outdoor temperatures of zero F.

SPECIAL CONTROLS MONITOR CLIMATE INDOORS FOR MICE

Guarded by a system of alarms and supersensitive thermostats, experimental mice at the Walker Laboratory of Sloan-Kettering Institute for Cancer Research live in rooms maintained at a temperature not allowed to vary more than two points from 74 F. Duplication of almost everything in the heating and cooling systems guards against shutdown from mechanical failure, and power is piped into the building from two separate and independent supply sources.

At the heart of the network of controls is a Supervisory Data Center, designed and built by Minneapolis-Honeywell Regulator Company. Equipped with a system of flashing red lights and alarm bells at the panel and strategically located points throughout the building, the center enables conditions in every part of the three-story building to be checked and regulated by pushbutton controls.

Forty-eight individual room thermostats, plus

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others regulating larger zones and areas, control the heating or cooling needed in each location. The data center gives the staff instantaneous reading on temperatures at 101 locations in the building. Besides temperature and humidity controls, the air in each of the animal rooms is completely replaced with fresh air, filtered through electronic air cleaners, eleven times each hour.

AIR CONDITIONING AIDS LABORATORY EFFICIENCY

Use of such highly delicate equipment as a balance sensitive to 0.000001 gm presented a problem with perspiration from hands of workers affecting the accuracy of measurements taken at the Micro-Chemical

Laboratory of Abbott Laboratories in North Chicago, Ill. Necessity for maintenance of a constant temperature and elimination of drafts and excessive humidity called for installation of an air conditioning system. Fourteen of Pyle-National's Multi-Vent Modular units were placed in the ceiling of the laboratory to provide even, low-velocity air conditioning. Shown is a section of the laboratory with one of the units, its diffusion plate removed, in the ceiling.

NATURAL GAS PIPELINE INSULATED

When Florida Power Corporation converted its St. Petersburg power plant to natural gas, consideration was given to low temperature insulation, temperature of gas flowing through the pipe lines almost always being lower than the dew point of the surrounding air. Basic objective in insulating the gas lines and associated fittings was to prevent formation of condensation between 32 F and ambient. Chosen for the job were two forms of a polyurethane foam material manufactured by Baldwin-Ehret-Hill, Inc., Gem-Foam

and Gemfil. Because of this material's low K-factor, 0.08 Btu/sq ft, hr, F at a mean temperature of −100 F, insulation thickness could be reduced greatly.

Gem-Foam, a pre-molded rigid polyurethane pipe insulating material, was applied to all the natural gas lines in a one-in. thickness throughout, molded half-sections being cut to fit pipes. Joints were pointed with an insulation sealer and the insulation covered with an aluminum jacketing.

Gemfil, which is poured in place, was cast around valves. An aluminum box was built around the fitting, the liquid poured through the top and allowed to foam. After curing, the top of the insulation was trimmed and an aluminum cover installed, the box thus serving both as a mold and finish covering.

AIR CONDITIONED LINER

Forty-five central air conditioning units, designed and manufactured by Thermotank Ltd., Glasgow, provide air conditioning for the 29,734-ton P & O Liner Arcadia. These units, made to measure for the spaces available, supply a total of 342,000 cfm of cooled air. Temperature and humidity of the conditioned air are regulated by automatic controls situated in various zones of the ship, affording fairly uniform cooling.

In maximum tropical conditions up to 100 ton of water will be extracted every day from the atmospheric air before it is circulated through the ship. Warm brine circulated to the same heat exchangers used for cooling provides heated air in cold weather. Viscous-type air filters have been incorporated in all main air conditioning units.

REFRIGERATED RAILROAD CARS

Centering around a 10-ton capacity compressor-condenser assembly and featuring completely automatic controls, the refrigeration system for 1000 of Pacific Fruit Express Company's 50-ft multi-purpose cars has been supplied by Trane Company. Diesel engines provide the power for the refrigeration units and are cited as being able to operate for 20 days without refueling. Electric heating is provided for cold winter months and for crossing mountainous regions. Constant temperatures ranging from zero to 70 F are maintained, with dehydration of fresh foods minimized by pinpoint temperature controls on the units.

HAIRPIN COILS FOR SKATING RINK

More than 10½ mile of 1¼-in. standard steel pipe produced by Jones and Laughlin Steel Corporation was welded into hairpin design coils for the refrigeration system of the Central Maine Youth Center ice rink in Lewiston, Me. After being tested, the coils were covered with loose sand and the 85 by 200-ft rink area was flooded and frozen, allowing engineers

to observe the performance of the system during an entire winter season before enclosing it permanently in concrete. Corrosion-inhibited calcium chloride brine is circulated through the system by two 20-hp pumps, each rated at 650 gpm. Refrigeration is obtained from two 75-hp direct-connected Worthington Corporation compressors. Supply and return headers are graduated using three to eight-in. pipe.

INSULATION FOR FREEZER ROOM

Construction of a freezer room for a Dover, Del., canner necessitated overcoming the above-normal heat and humidity common to this area. Armstrong Cork Company's Armaflex, a flexible foamed plastics insulation, was used on the cold lines and Armalite, a foamed plastics insulation board, was used for the walls, ceiling and floor of the freezer room itself and the anteroom in front of the freezer room.

Two layers of ¾-in. sheet Armaflex, held in place and sealed at the joints and seams with waterproof adhesive, covered 3½-in. lines and tanks, smaller lines being insulated with slip-on pipe covering. Oddly-shaped controls and flanges were covered with scraps of the insulation, built up to form an even surface, and then covered with larger sheets.

Designed to operate at –35 F, the freezer room is divided into two cells, separated by a 10-ft long, 6-in. thick self-supporting wall of Armalite. Outside brick walls were insulated with 9-in. Armalite while the ceiling and floor were insulated with 10 in. of the material.

INFRA-RED HEATS FACTORY

Estimated heat losses of 3,640,000 Btu/hr in Solem Machine Company's Rockford, Ill., plant were offset by installing 82 Perfection Industries' infra-red heaters. Installed at heights of 26, 28 and 42 ft in the high-ceilinged plant, the units have a total installed capacity of 3,888,000 Btu/hr. Most of the heaters were placed along the building perimeter where the largest heat losses occur.

Four sources, each with a capacity of 12,000 Btu/hr, are combined into these units with a type of parabolic reflector that directs the radiation downward. In this installation, the heaters are angled slightly to the horizontal so that infra-red radiation covers more floor surface without concentration of the heat energy on any specific area.

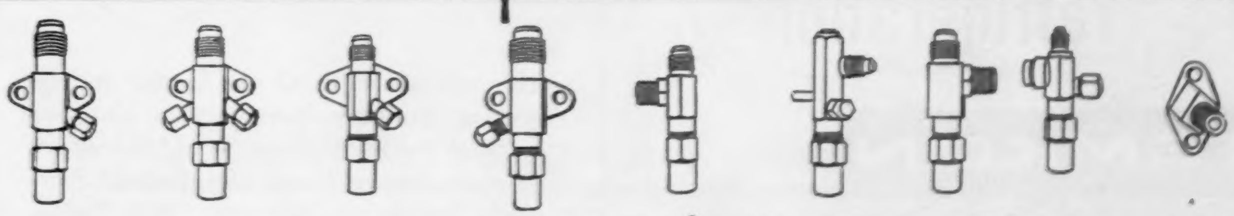
DRIVE-IN RESTAURANT PROVIDES COOLING FOR PATRONS

Placed in car windows by a carhop girl, flexible hose pours cold air into autos, keeping motorists cool while parked at a North Carolina drive-in restaurant. Cool air, available for 14 cars at one time, is produced by a 30-hp Westinghouse air conditioning unit. Underground ducts bring the air to the parking lot posts, each equipped with two hoses. A damper in the hose outlet permits motorists to adjust the amount of cold air they receive.

MUELLER BRASS CO.

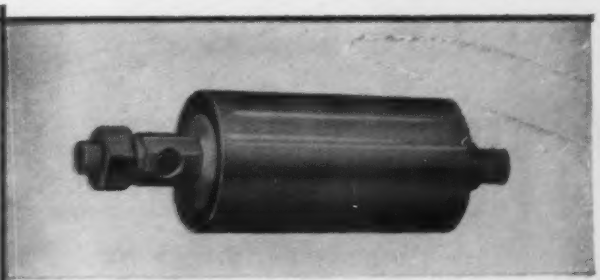
introduces . . . hydrogen brazed steel compressor valves that combine high quality with low price

A new line of Streamline® Hydrogen brazed steel compressor valves is now available to you at amazingly economical prices. They incorporate the same high quality construction as the forged brass valves developed and introduced by The Mueller Brass Co. many years ago. The combination of steel construction, and new Hydrogen Brazing techniques has made possible the production of sturdy, dependable valves that will give outstanding performance in your products, yet hold down those all-important costs.
Included in this new line of valves are compressor valves, (single or double port and adjustable), steel evaporator liquid valves and receiver valves. All are available in both flare and solder type connections. Write us for full information on cost and engineering specifications . . . a Mueller Brass Co. Sales Representative will be glad to call on you.



A NEW COMBINATION . . .

Mueller Brass Co.'s new exclusive combination filter-drier and liquid indicator assembly is also a hydrogen-brazed all steel product. It combines the perfectly balanced DRYMASTER Filter-Drier and the easy-to-read SIGHTMASTER liquid indicator into one economical, compact, quickly installed unit.



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MADE IN U.S.A. TO THE STANDARDS OF AMERICAN INDUSTRY

MUELLER BRASS CO. PORT HURON 15, MICHIGAN

BULLETINS

(*Continued from page* 118)

tions and major features are included.
H. K. Porter Company, Inc., Forge and Fittings Div, Roselle, N. J.

Automatic Soldering Guide. Offering data on solder preforms, their selection and use, conversion to automatic soldering and flux-filled washers for this process, the "Guide to Automatic Soldering" also includes technical information on solder alloy and flux selection. Information on allied problems is given in two other bulletins, "Tips on Soldering" and "Flux-Finder Guide."
Alpha Metals, Inc., 56 Water St., Jersey City 4, N. J.

Exhaust Ventilators. Two flyers, VEV-58 for a vertical model and HEV-59 for a low silhouette hooded model, contain engineering data and product descriptions for two of this manufacturer's line of roof ventilators. Designed for both commerical and industrial application, the hooded exhaust ventilator has a 360-deg near horizontal discharge pattern cited as providing maximum efficiency regardless of wind direction or velocity. A high-velocity discharge on the vertical unit exhausts fumes, smoke, dust-laden and foul air high above the roof to prevent re-entry to the building. When the fan is operating, butterfly dampers blow open to permit free exhaust of the air; when the fan is shut off, the dampers immediately close.
Western Engineering and Manufacturing Company, 4114 Glencoe Ave., Venice, Calif.

Gauge Mount. Damping mechanical vibration by isolating the pressure gauge from the compressor or piping system with a rubber damper, the Stedy-Mount gauge mount is cited as lengthening gauge life. Designed for bottom-inlet gauges weighing up to two lb, the mount, described in Flyer 835, can be used for any gas or fluid that does not attack carbon steel.
Vilter Manufacturing Company, Milwaukee 7, Wisc.

Heating and Cooling Handbook. Recommended for engineers, architects, builders, contractors and dealers interested in zone control hot water baseboard heating, a 70-page design handbook provides sections on construction, installation, dimensions, specifications and features. Covered in this comprehensive reference book is the complete system: compact boiler-burner, zone control valves and baseboard radiation elements.
Edwards Engineering Corporation, 1 Alexander Ave., Pompton Plains, N. J.

Control Drives. Included in revised 6-page Product Specification P81-1 is a new high torque control drive. All four sizes of drive, producing torques from 50 to 2250 ft lb, and suitable for special high temperature applications to 250 F, are described.
Bailey Meter Company, 1050 Ivanhoe Rd., Cleveland 10, Ohio.

Water in Air Conditioning and Refrigerating Systems. In question and answer form, this flyer explores use of water in cooling systems, indicating how operating cost of a system may be minimized through due consideration being given to the functioning of the water. Some questions asked are: "What are the operating troubles that are traceable to the water?", "Why is the circulating water in cooling towers, air washers and evaporative condensers more corrosive than ordinary water?", and "What causes scale formation?".
Water Service Laboratories, Inc., 423 W. 126th St., New York 27, N. Y.